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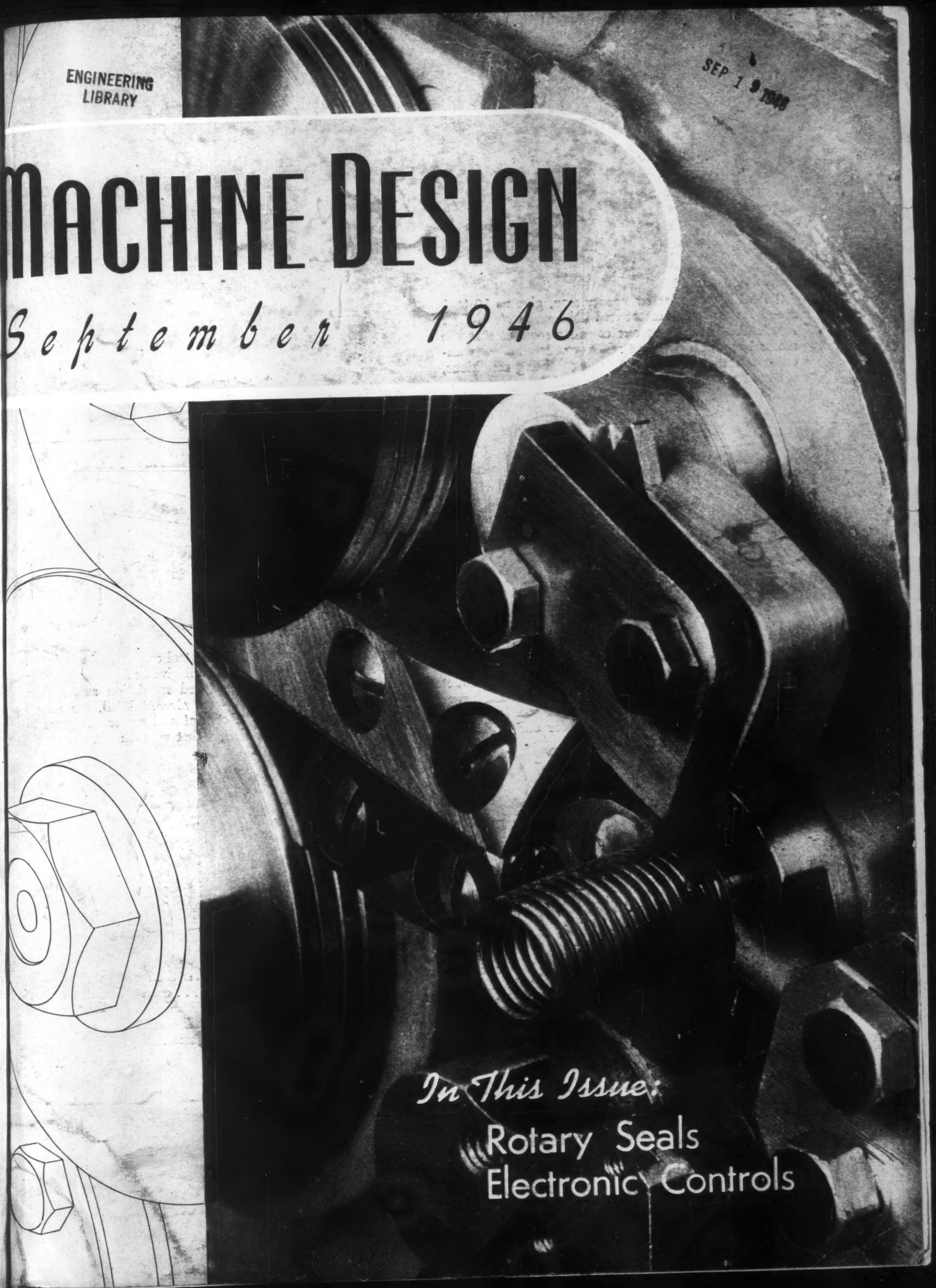
MACHINE DESIGN

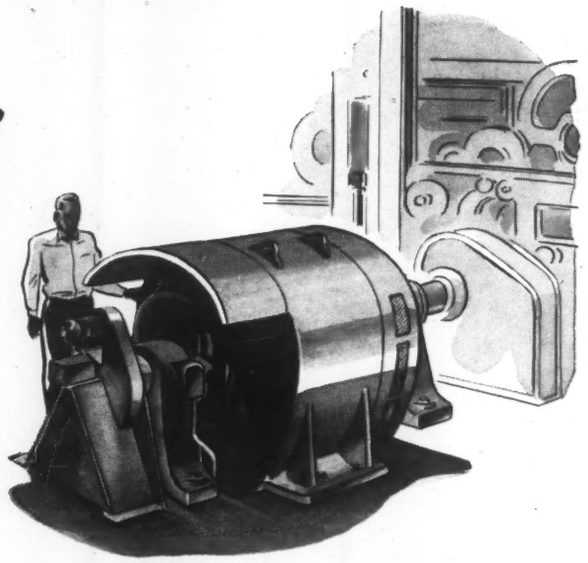
September 1946



In This Issue:

Rotary Seals
Electronic Controls

"What! Stop a 30" Calender Roll in Less than 14" of Roll Travel!"



"Right! And besides that emergency stop, give us 10 to 1 speed range on the motor and hold speed *close*. The torque characteristic of our calender requires that the motor have high overload capacity at low speeds without overheating. And because of operating conditions, we want sparkless commutation too!"

B-b-u-t-t... "No buts" said Mr. Rubber Mill Man, "that's *exactly* what we need. You see, we're experimenting with new materials on our rolls, and since we can't *pre*-determine best roll speeds for them, we must have a drive that'll give us complete *range* of speeds, and at the same time, complete *protection* for our workers".

"In A Nutshell, you and the control engineers must design a calender drive system that'll give us *higher* and *lower* calender speeds, *closer* speed regulation and *sudden* emergency stopping — *all three!*" Well, we set up meetings between the control and motor designers to plan a *coordinated* design that would meet the needs.

An Automatic Speed Control incorporating both our m-g set and "Regulex" exciter was designed to hold selected speed within close limits. And, after a careful analysis of required load characteristics, we designed a 400 hp d-c motor of low inertia for quick stopping . . . and which also provided speed range from 25 to 250 rpm.

Remember That 14" Stop? After the system went in, we stopped the rolls first at 18" . . . easy. Then at 14" . . . at 12" . . . at 9"! And we probably would have stopped 'em even shorter but for possible danger of too sudden stops to the calender gear system. And commutation? After 6 months, the operators say it's perfect. A 2048

Moral: Every time Allis-Chalmers discovers new ways of solving *special* motor problems, like this one, it also learns how to build better *standard* motors for you! Watch for these new and better motors from A-C. ALLIS-CHALMERS, MILWAUKEE 1, WISCONSIN.

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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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THIS MONTH'S COVER: Tooling of spring coiler designed by the Torrington Manufacturing Co. Coiling principle employed—pushing wire around an arbor against a coiling point—is fast and accurate. Coil diameter of spring is 7/8-inch, using 0.072 inch wire.

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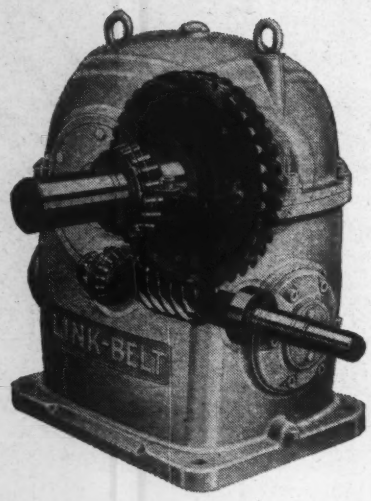

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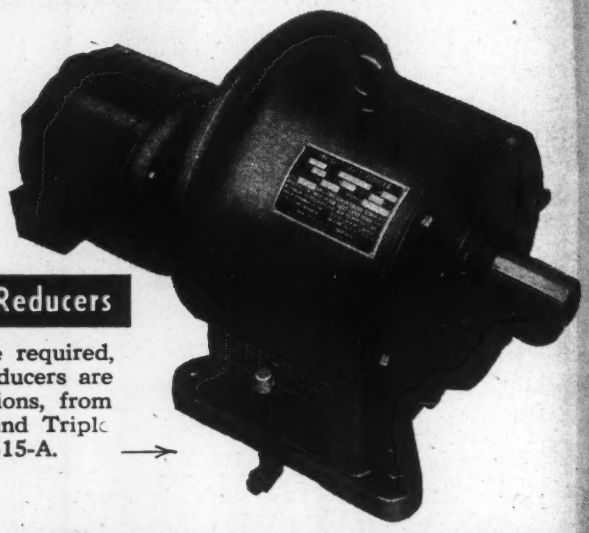

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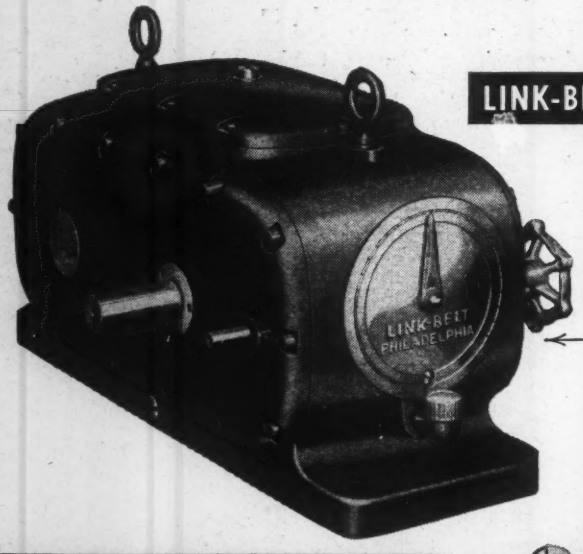

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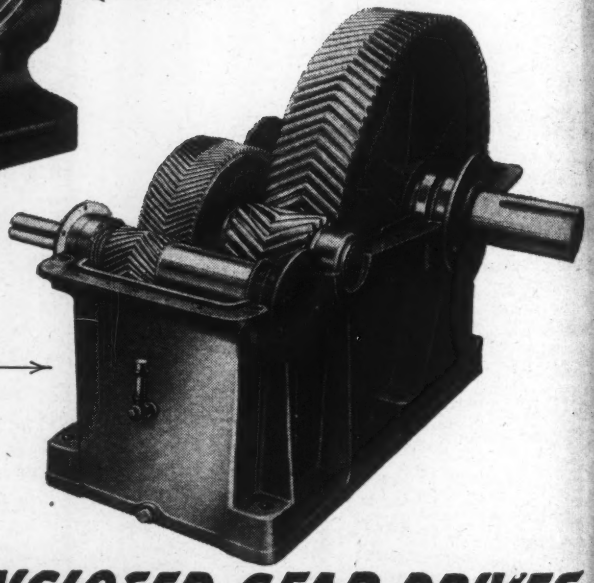

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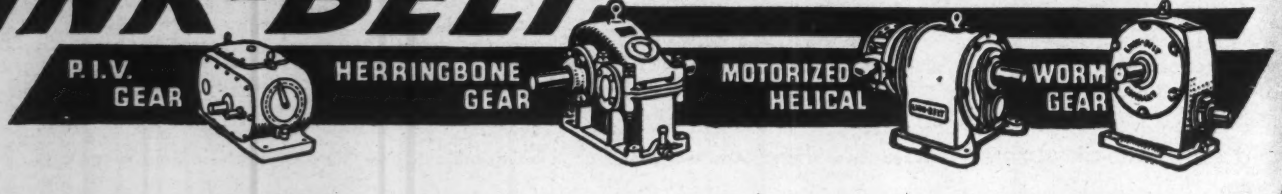

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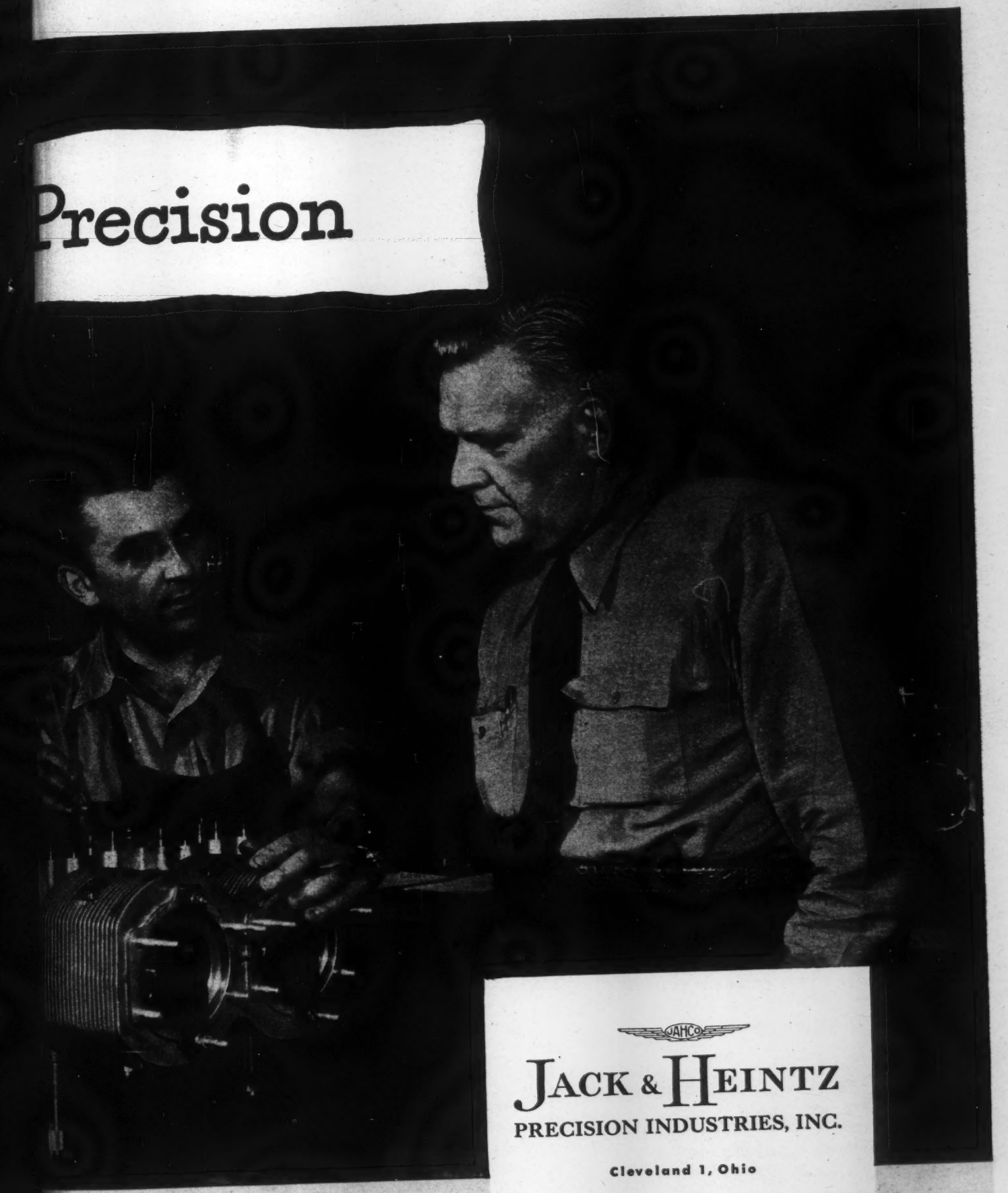

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ELECTRONIC GAUGES AND MAGNETOS

Topics

MAGNETIC GAGE for measuring the thickness of thin coatings on steel utilizes a solenoid with a soft, movable iron core and an indicator scale. Critical distance from the coated steel at which the solenoid attraction for the core will just overcome that of the steel is a measure of the thickness of the coating. Maximum error is ±5 per cent for 0.002-inch films and ±3 per cent for thicker coatings.

IRON POWDER prepared direct by reducing pulverized ore at comparatively low temperatures has been developed in Germany by the Kaiser Wilhelm Institut fuer Eisenforschung. Reduction is carried out in several stages, allowing removal of impurities between stages.

CEMENTED CARBIDE bushings, used as fairleads for aircraft target towing cables, have guided more than 3,000,000 feet of stainless steel cable without showing signs of wear. Because of the high speed at which the cable is paid out, other bushings are destroyed rapidly.

PUBLIC ENTHUSIASM for new and radical designs should not be misinterpreted as a desire for possession, warns R. E. Bingman, industrial designer. He says man is comfortable and secure in the presence of the familiar although adroit design may lead him farther than he realizes.

URANIUM, source of energy for the atomic bomb, was formerly used to make green dyes for porcelain enamel. Because of the scarcity of this metal, suitable dye substitutes are being sought.

ELECTROLYTIC METAL POLISHING technique for surfaces that do not lend themselves to tank treatment gives more rapid results, prevents "flowed" layers and requires less skill than mechanical polishing. A portable brush electrode is used which consists of a metallic core enveloped in a solution-retentive dielectric material. Developed at Frankford Arsenal the method involves moistening the brush in a suitable electrolyte, applying direct-current power and gently pressing the brush against the object to be polished.

PACKAGED MANUFACTURING PLANTS are being developed for export to foreign manufacturers. Complete production units, they include buildings and all machinery prefabricated and ready for erection as well as services of engineers for putting the plants into operation, training labor and consulting.

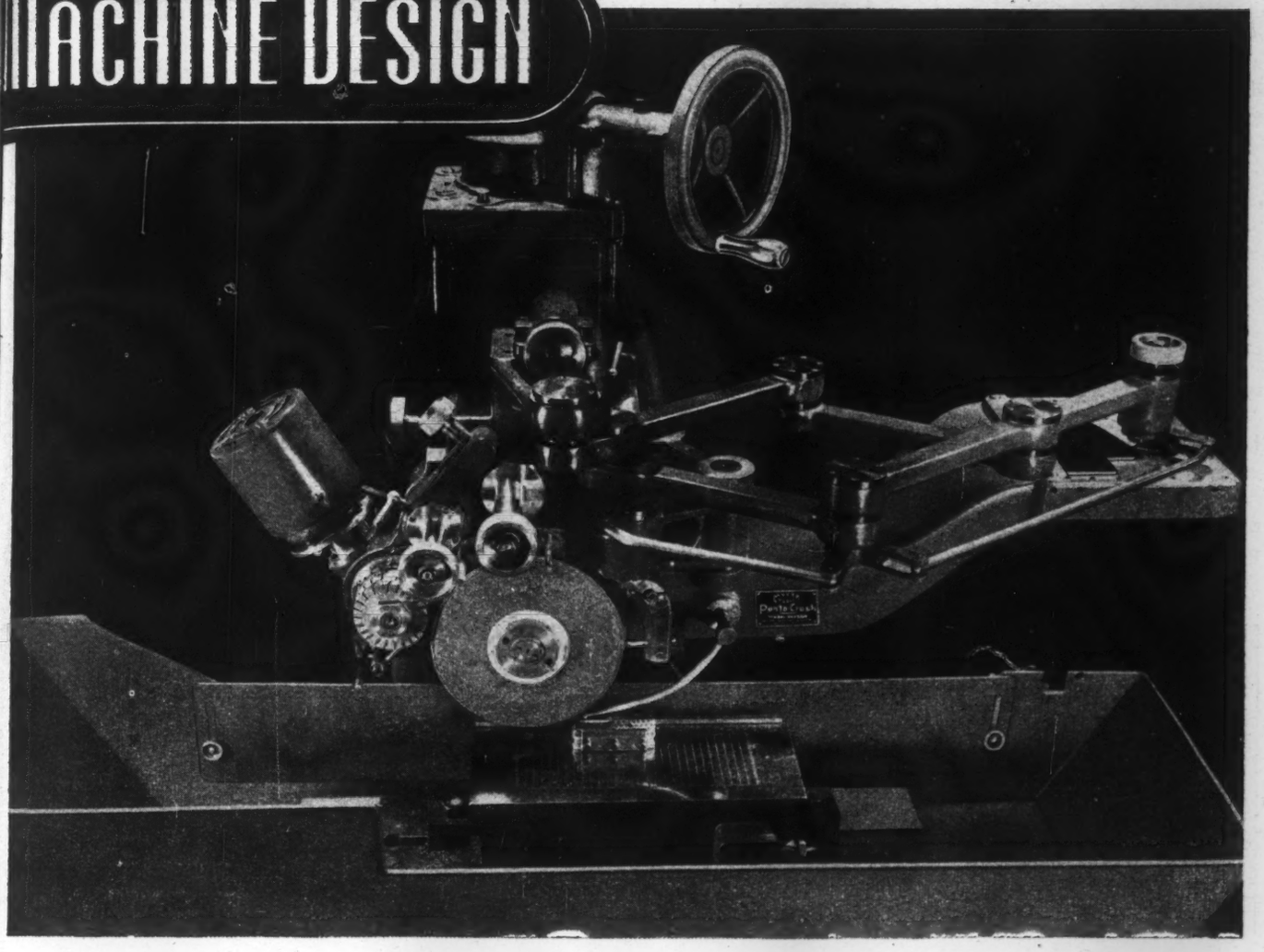
BEARING MATERIALS composed of small portions of tin, nickel and copper with the rest commercially pure aluminum have been reported to give superior performance under extreme loads at high temperatures. These bearings resist damage by particles of bronze and steel, which they absorb, and show high resistance to corrosive attack from lubricants. Addition of silicon improves antifrictional and antiscoring properties.

HIGH-STRENGTH ALUMINUM alloys of the aluminum-manganese-zinc-copper type recently perfected in this country were found to have been used extensively in captured Jap Zero planes.

NATIONAL DISTRIBUTION COUNCIL has been created by the Department of Commerce to help industry improve its distribution capacity and efficiency, realizing that the present seller's market may soon become a buyer's market.

TRANSPORT PLANES each powered by two G. E. propjet units, are expected to travel one mile on a gallon of kerosene compared to one and one-half miles on a gallon of gasoline now obtained with standard engines. Cruising at 350 miles an hour, net savings are expected from lower cost of fuel and more revenue miles per day.

MACHINE DESIGN



Combination Grinding Wheel Dresser

. . . features pantograph for guiding diamond—also self-driven crush-former

By J. Robert Moore
Moore Special Tool Co. Inc.
Bridgeport, Conn.

DIAMOND dressing and crush forming of surface grinder wheels generally call for the use of separate devices requiring considerable set-up and operating skill, particularly where odd contours other than true radii and straight surfaces are involved.

However, with the development of the unit shown in the lead illustra-

tion, known as the Moore Panto-Crush wheel dresser, both crush forming and diamond dressing were accomplished in a single mechanism. This combination unit, permanently mounted on the wheel spindle of a surface grinder, employs a precision-built pantograph for guiding the diamond from a template, and a self-driven crush former.

Pantograph of the unit, *Fig.* 1, has a 10:1 ratio between the movements of the stylus and the diamond dressing tool. This ratio was chosen as being optimum for the size shown, to be used with a grinding wheel approximately 7 inches in diameter by ¾-inch wide. The stylus and diamond-holding arms are angularly articulated, that is, capable of co-ordinated angular rotation in addition to the conventional pantograph movements. Contact between the stylus and the template is maintained by merely holding the stylus against the template as the form is traversed. Accuracy of form of the template can be held to plus or minus 0.001-inch and the 10:1 ratio reduces this to 0.0001-inch on the wheel and workpiece.

In addition to form generation, a method is necessary for feeding the diamond into the wheel so that as the wheel wears the form may be duplicated on a progressively smaller wheel. This could be accomplished either by moving the template on slides or by moving the pantograph main pivot. Using the first method, infeed of the template would, of course, permit the stylus and diamond to approach the grinding wheel. However, since this is at the large end of the 10:1 ratio, it would necessitate very large and bulky slides; to infeed the diamond one inch would require a 10-inch table travel at the outboard end. The required weight would apply excessive overhang and unbalance to the entire device.

Accordingly, the second method was employed, *Figs.* 1 and 2. The main pivot is supported rigidly by rectilinear dovetail slides and ways which are hardened, ground and lapped, and is mounted in a central position over the spindle, thus reducing overhang and unbalance. Slide ways are actuated both radially and axially by lead screws fitted with dials reading to 0.001-inch (see *Fig.* 2).

The diamond arm is not directly supported by the pantograph mechanism, but by a rigid knuckle or main supporting link as shown in *Fig.* 1. The pantograph merely serves to move this knuckle around, without permitting up or down movement from inadvertent up or down thrusts the operator might apply to the stylus arm.

Weight and overhang of the pantograph are considerable and cause a certain amount of sag in the whole mechanism. If the diamond arm were carried directly upon it,

Fig. 1—Pantograph has 10:1 ratio between stylus and diamond. Sections show design details of primary pivots

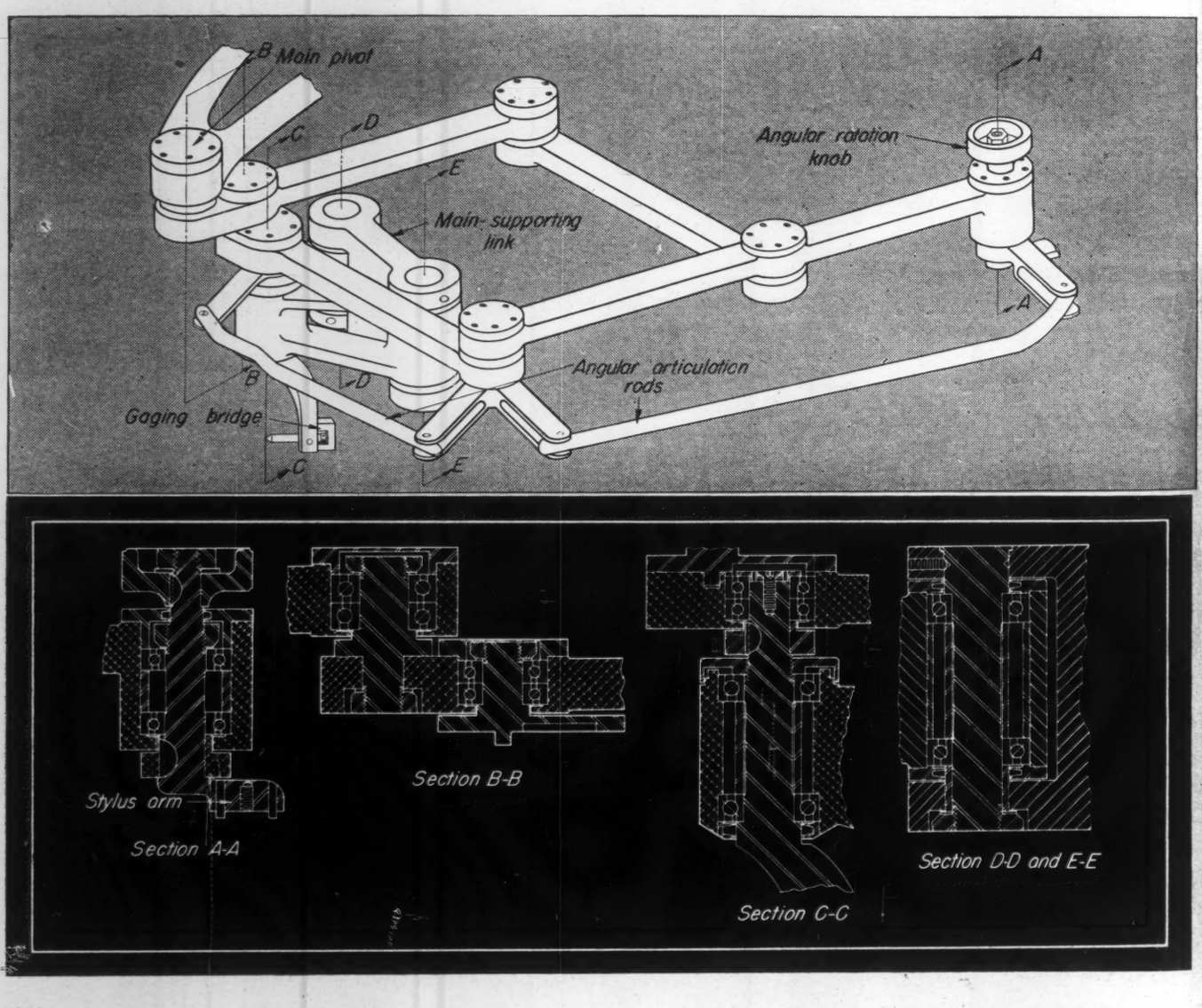


Fig. 2—Right—Design details of dovetail slides and ways which support main pivot of pantograph and permit adjustment of its position in reference to the grinding wheel

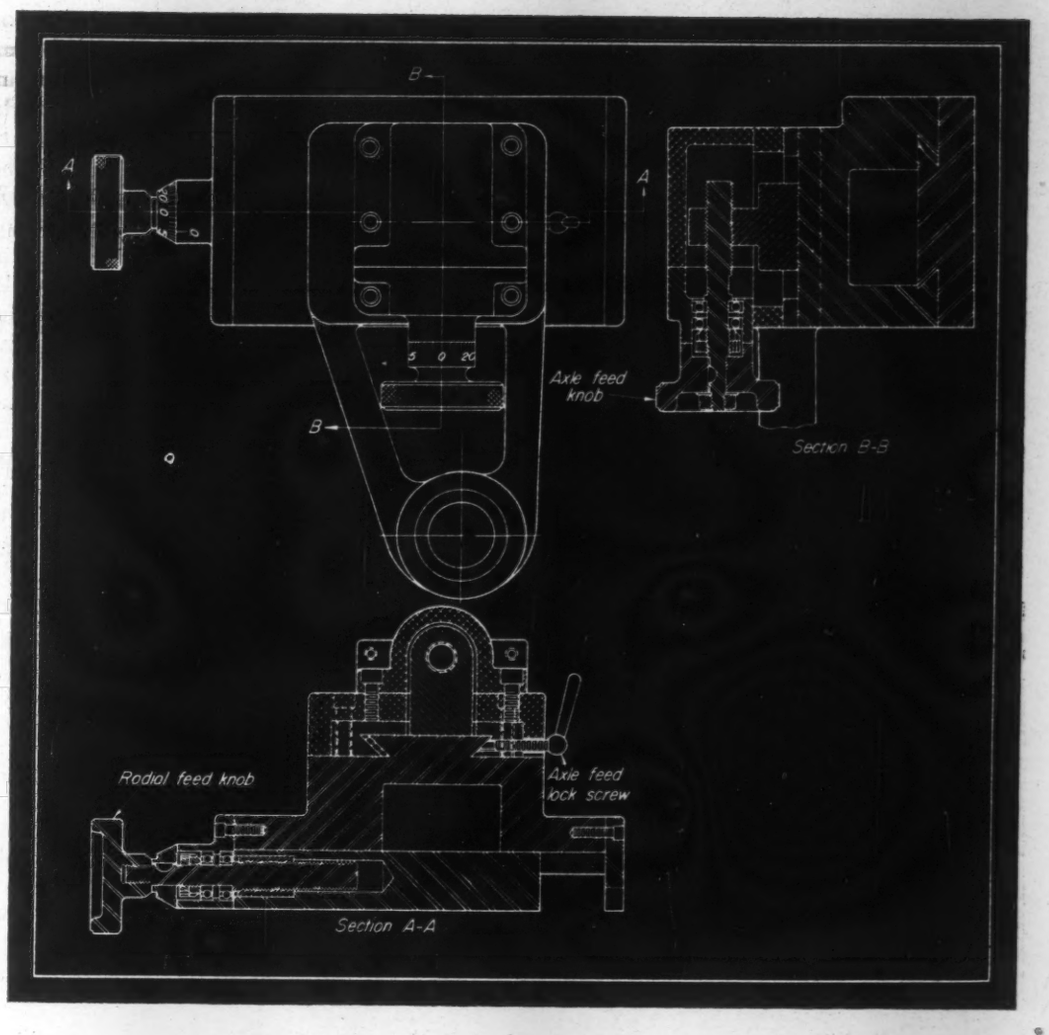
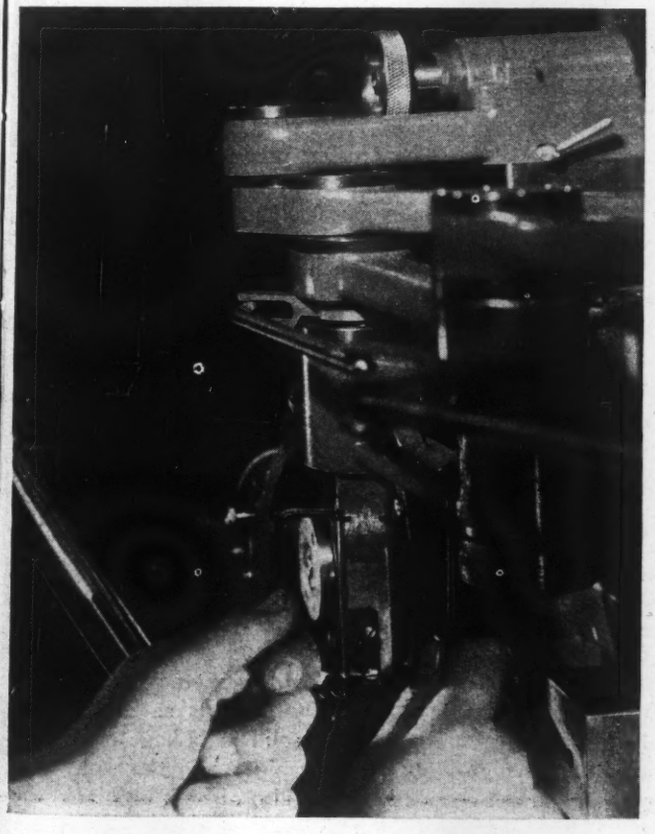


Fig. 3 — Below — By tightening pipe plug on one side and loosening plug on other side a proportionate amount, diamond-holding arm is sprung to align diamond with arm pivot



it would give the diamond a tendency to dig into the wheel. This tendency is prevented by two devices:

1. By casting the pantograph arms of aluminum alloy No. 35 HT, which has optimum modulus of elasticity in relation to its weight
2. By the relative locations of the main pivot and the knuckle. These are so arranged that, while downward thrusts of the stylus arm tend to push the diamond into the wheel, resultant deflection of the knuckle is equal and opposite, neutralizing the effect of up or down movements of the stylus arm.

A difficulty encountered in most conventional tangent-to-radius dressers is that of correctly relating some known point on the diamond to the center of rotation of the radius device. This relationship must be established by cut-and-try methods.

It was felt that a means should be devised whereby the stylus could be accurately related to the diamond by direct measurement. To do this, two approaches were possible:

1. Accept the random shapes of the diamonds and make a special stylus to match each diamond
2. Standardize shapes and sizes of both diamonds and styli, thus permitting complete interchangeability.

The first approach was discarded because of the high cost and nuisance of continually making special styli, and because of the fact that a small amount of wear would quickly render the diamond useless. Styli and diamonds, therefore, are made to standard matching radii. The radius was chosen as the standard shape because of its

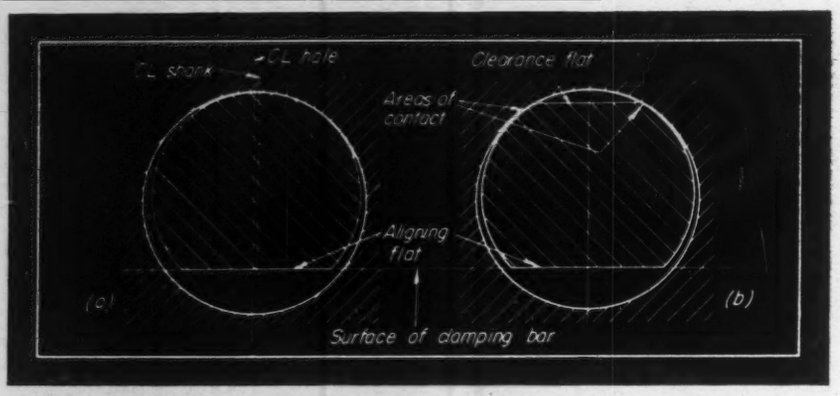
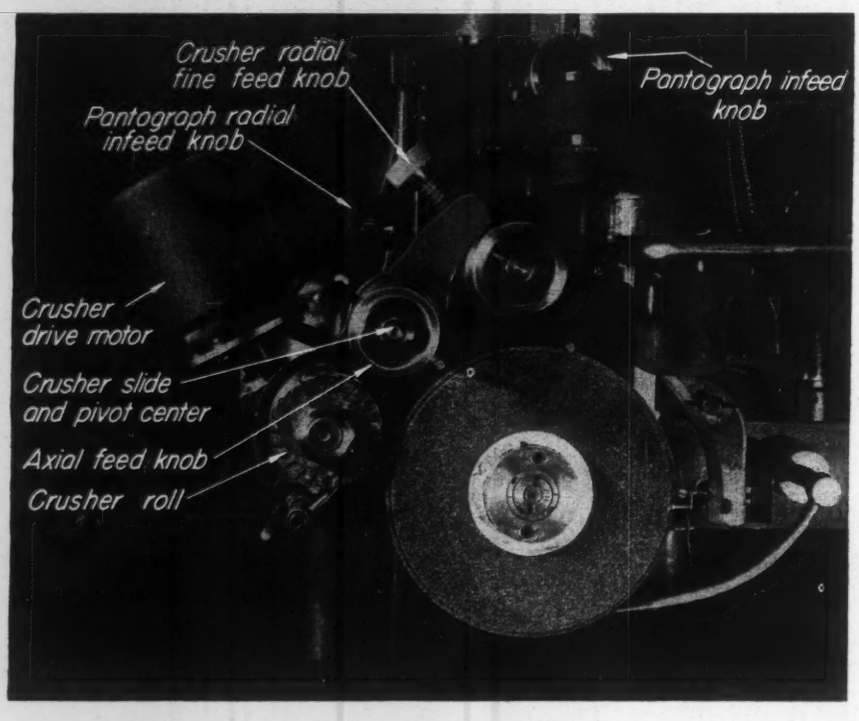


Fig. 4—Perfectly round diamond shank, as at (a), permits slight eccentricity. Flat on shank, as at (b), gives two-point contact and prevents eccentricity

Fig. 5—Below—Crusher-roll unit with primary parts identified. This unit is self-contained, having its own drive motor and positioning controls



strength and easy measurability. Since the pantograph ratio is 10:1, the stylus must be the same shape as the diamond, but 10 times as large. Angular articulation of the stylus and diamond arms adds greatly to the practicability and economy of this system, since the sides of the diamond, as well as the front, may be used for dressing. In this manner, the diamond may be worn to a new, constantly-increasing radius. Substitution of the next larger stylus in the same series and advancing the diamond to match it permits continued use of the diamond.

To achieve complete interchangeability, high precision is required in holding the center of the diamond radius central with its shank, in holding the shank central with the diamond arm, and in holding the stylus central with the stylus arm. The diamond shank is centralized in a carbide bushing, and the hardened, ground stylus locates on pins in jig-ground holes.

Position of the bushing in the diamond-holding arm relative to the arm pivot center is adjustable by means of a pair of opposing pipe plugs, which serve to bend the arm sidewise for any slight adjustment required. This adjustment is factory set, and not intended to be disturbed by the user. *Fig.* 3 shows how this adjustment is made.

During the development of the diamond-holding end of the arm, the question arose as to whether the diamond shank should be held in a Vee or in a round bushing hole. The round hole has the advantage that it prevents insertion of anything but a perfectly clean, nick-free shank. It has the disadvantage, of course, that a "tenth" or two of looseness is required to insert it, with the danger of it being held eccentrically, *Fig.* 4*a*. The Vee would appear to offer the only way to insure contact with the sides, and thereby maintain centrality. This method has the disadvantage, however, that nicks or dirt can get between the contacting surfaces, and disturb location.

Compromise Plan Adopted

A compromise plan, *Fig.* 4*b* was arrived at, whereby the advantages of both methods were achieved. A small clearance flat, on the round shank, prevents contact with the hole on the side opposite the clamping flat. This shifts the contact to two portions of the hole about 90 degrees apart. Thus, the action of the Vee is simulated in that the effect is a two-point contact, yet the advantage of the clean round hole—preventing insertion of the shank if nicked or dirty—is retained. The bushing, being tungsten carbide, is practically indestructible.

The crusher roll unit, *Figs.* 5 and 6, is power driven and drives the wheel spindle by friction. Whether the roll should drive the wheel or vice versa is simply a question of expediency. Either method appears to work equally well provided the roll is not permitted to idle on the wheel. Sufficient pressure is required to make sure there is no drag, in which case the wheel would start to grind the roller. Speed of rotation is not felt to be critical. A 2⅜-inch diameter roller operates against a 7-inch wheel, the crusher rotating at 80 rpm.

There are two good reasons for power-driving the crusher roll in this unit. One is to eliminate the necessity for an auxiliary slow-speed drive for the grinding spindle. The other is that the crusher roll arbor may be employed also as an arbor for grinding the crusher roll. This eliminates the necessity for an expensive auxiliary machine for fabricating the roller.

Manufacture of the roller by this method is, in fact, extremely economical. Rolls with forms as deep as ¾-inch may be ground from the solid in ¾-hour or less by plunging the grinding wheel rapidly into the roll. Not only is the roll economical, it is also of high accuracy, the

form being the exact inverse of a diamond-dressed, precisely formed wheel. An additional advantage is that circular form tools or other circular pieces also may be ground on the crusher roll arbor.

An interesting design is the slide bar, carrying the crusher arm, *Fig.* 6. Motions in two directions are necessary, i.e., axially, to orient the crusher to the position of the form on the wheel, and radially, for infeed of the roll against the wheel, to permit grinding the roll and for the roll-crushing operation.

Slides Too Bulky

At first, a pair of rectangular slides was conceived for this, one of the slides traveling axially and the other radially. However, space and expense considerations led to a search for some other method. The final solution comprises a round bar, hardened, ground and lapped, supporting the crusher arm, sliding in two hardened and ground bushings, also line-lapped. This method provides ample rigidity, yet moves smoothly in both desired directions by sliding or turning.

The crusher is fed into the wheel by means of an eccentric cam, turned by hand, *Fig.* 6. This cam provides for quick infeed and quick withdrawal, and prevents the roll from dwelling too long on the wheel. A slow, gradual approach would wear the roll unduly. Further adjustment is provided by the easily-accessible adjusting screw, labeled "radial fine feed knob" in *Fig.* 6.

Axial movement is controlled by a lead screw, with dial graduated to correspond with that controlling axial infeed of the pantograph. Since the pantograph ratio is 10:1, the main pantograph pivot moves 1/9 farther than the diamond. Automatic compensation for this difference is achieved by providing 9-pitch lead screws for the pantograph slides, and a 10-pitch screw for the crusher. All dials are graduated to read in thousandths of an inch, in terms of actual movement of diamond or crusher.

Experience up to now indicates that the crusher roll operates most satisfactorily if serrated angularly, as shown in the illustrations. If left unserrated, crushed particles remain between the roll and wheel, and interfere with successful crushing. Serrations are angular to reduce the bumping effect each time a slot is passed. Spaced unevenly, they cannot "track," regardless of relative diameters of wheel and roll. A spout is arranged to direct a stream of coolant at the juncture of the crusher roll and

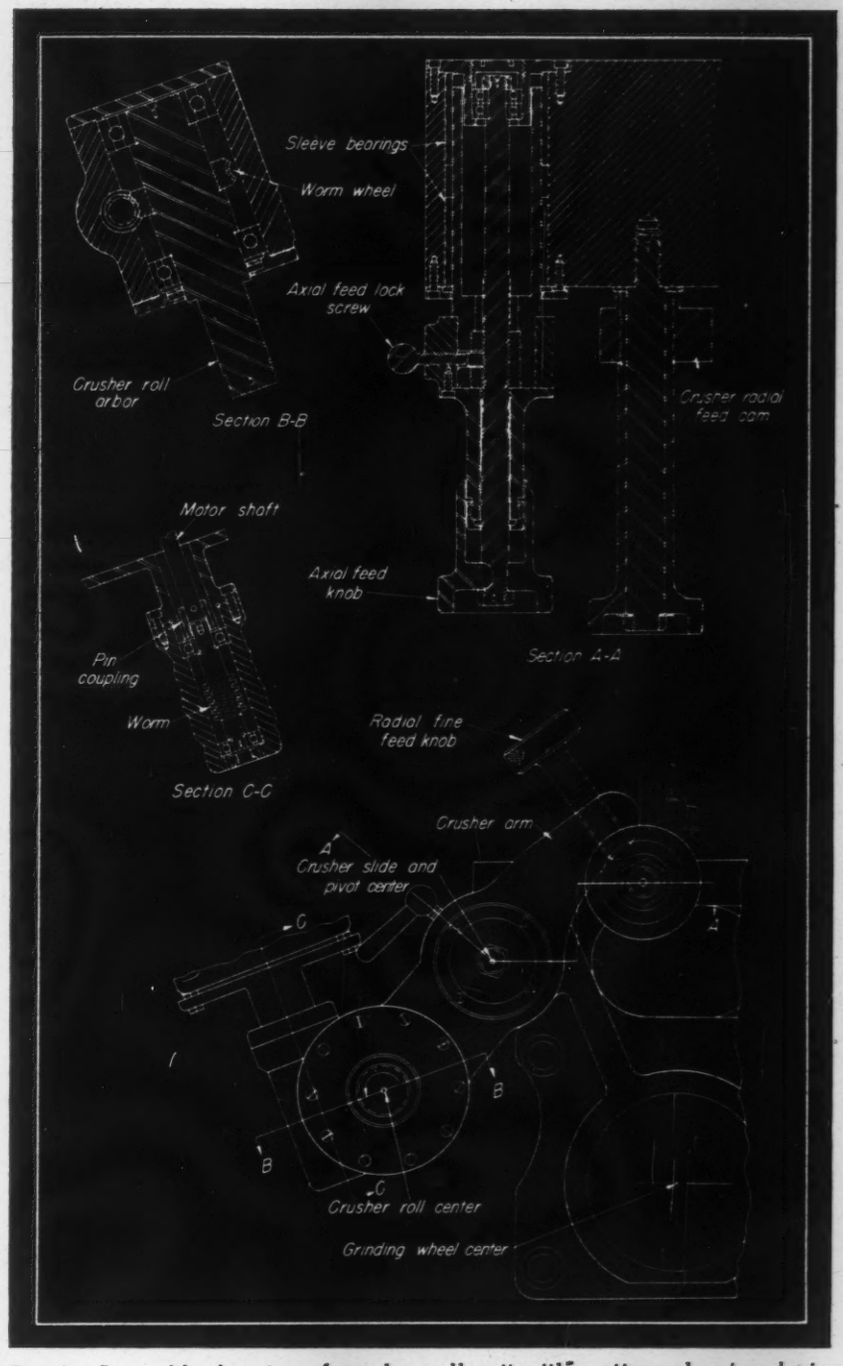


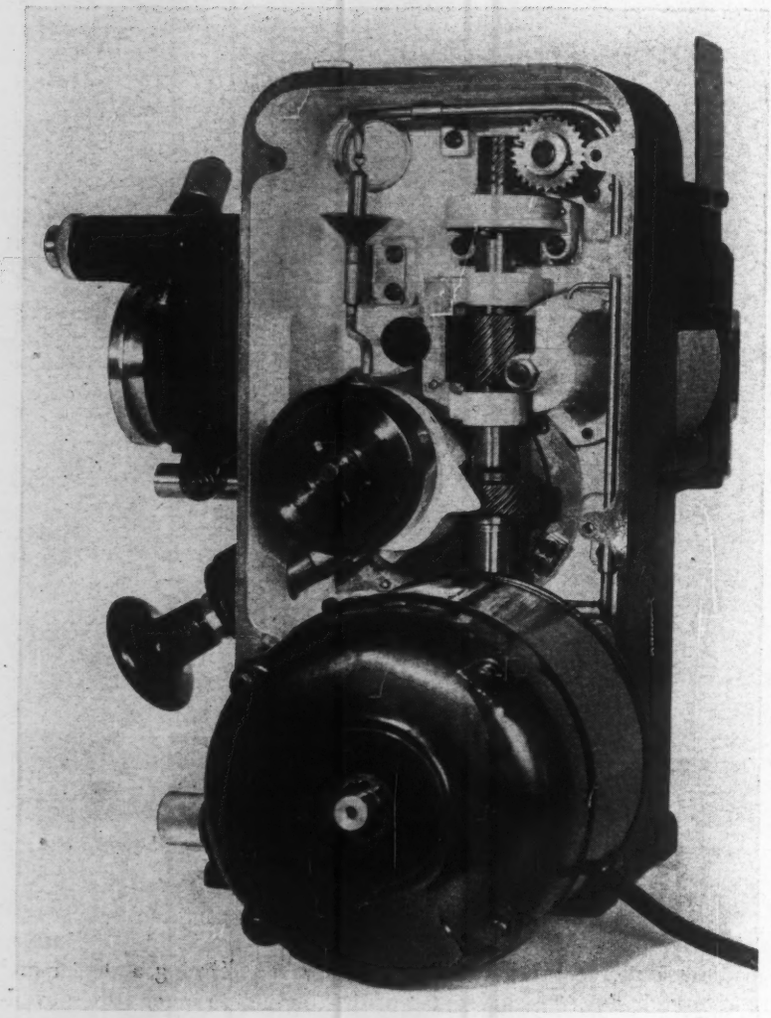
Fig. 6—Assembly drawing of crusher-roll unit with sections showing design details of various important components

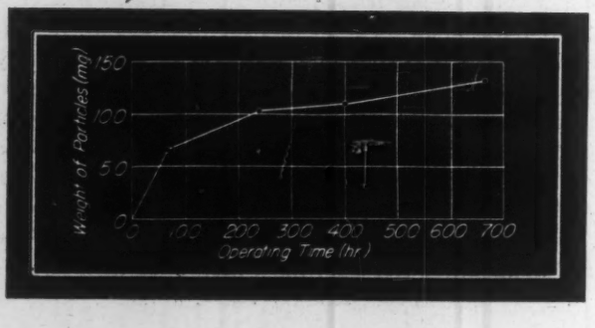
the wheel, to wash away the crushed particles.

Advantages offered by the combination of crush forming and diamond dressing in a single device include: (1) Either method in immediately available for any given job; (2) skill is required only to make the template, from which diamond dressing becomes a simple, repetitive operation; (3) irregular contours can be generated which are impossible to produce with conventional tangent-to-radius dressers; (4) optimum accuracy and economy are achieved in making crusher rolls and in using them; (5) switching from one method to the other does not disturb the setting of the workpiece or lose the location of either truing device with respect to the wheel or workpiece; (6) wheel spindle can be raised or lowered without disturbing setting of either truing device.

In addition, there is also the advantage of optimum economy in diamond consumption, the diamond being used only to dress the wheel for grinding the crusher roll and for finish grinding the workpiece.

Scanning THE FIELD for Ideas





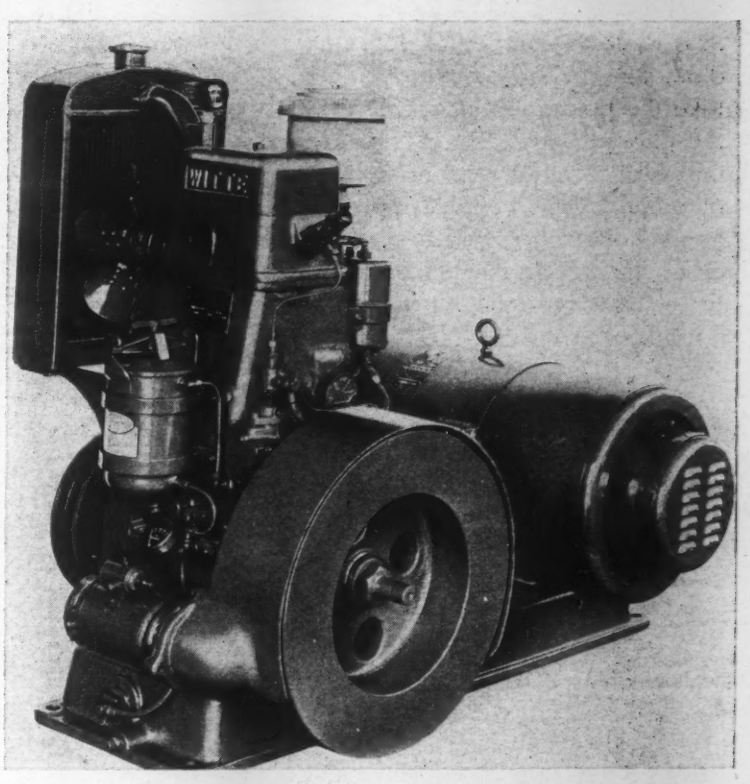
MAGNETIC oil filter to remove contaminents such as iron dust from the lubricating system for a delicate precision mechanism is shown in the illustration at left. Installed in a Philips Co. (Holland) 35-mm projector, the filter may be seen suspended in the funnel-shaped reservoir feeding a geneva mechanism. Oil to this mechanism flows over the strong magnet in such a way that the iron particles are attracted chiefly to the convex pole and neither restriction to flow nor extreme washing or scrubbing action are developed as the particles build up on the filter.

All shaft and bearing bushings are lapped or honed; the maltese cross in the geneva mechanism is finished by a special lapping treatment to give smoothness and precision within 2 microns. Gears, however, have an ordinary finish obtained by hobbing with a fine milling cutter because successful grinding or lapping methods have not yet been developed for such small gears.

Of the oil passing through the pumps only that portion is filtered which lubricates the maltese cross. Only magnetic particles are captured; particles of bronze or white metal remain in the oil. Fortunately the parts which are subject to initial wear are iron or steel such as gears, cast-iron bearings, etc. Length of captured particles seems to lie between 0.4 and 4 microns. Since the circulation is rapid, all the oil eventually finds its way to the filter.

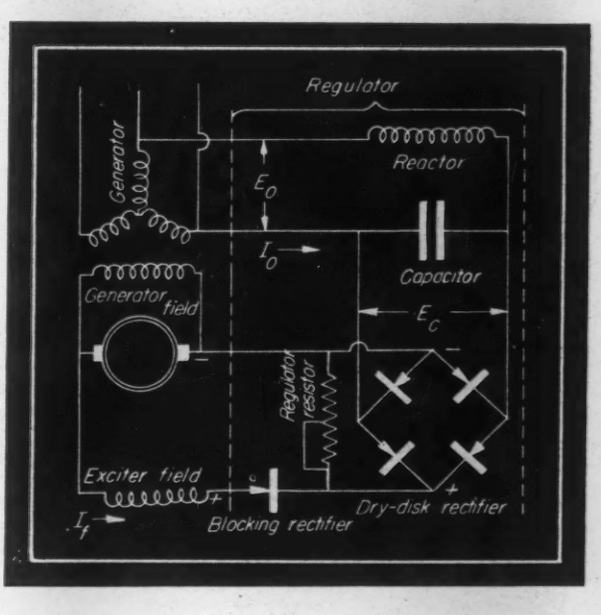
Collection of particles decreases rapidly after the initial working in as may be seen in the graph at left. This curve shows the wear and tear on a projector in terms of weight of iron particles deposited on the magnetic filter in terms of hours of operation. In the maltese cross, wear and tear should be held to a minimum. Play due to wear causes the unit to knock harder and then wear off even more rapidly.

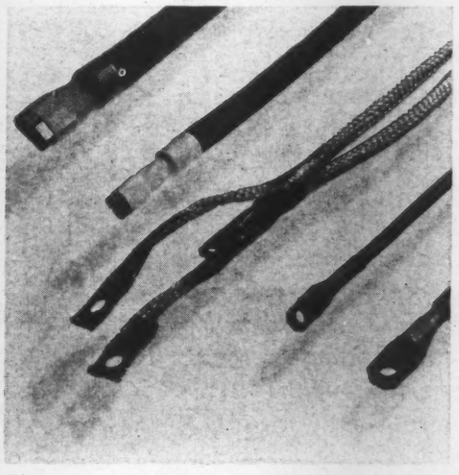
Constant voltage from the diesel generator shown at right is achieved automatically from a built-in regulating circuit without use of rheostat or external operatng instruments of any kind. Shown schematically at right below is the regulating circuit for the generator designed by the Witte Engine Works. Operation is based upon the principle of the amplification obtained when alternating-current voltage from the generator is supplied to a near-resonant circuit consisting of a reactor and capacitor in series. This series circuit is so designed that a resonant condition is developed which allows the voltage across the reactor and capacitor to be much higher than the voltage applied to the series circuit. The amplified voltage is used to control the voltage of the exciter and thus regulate the generator.



As seen in the diagram the voltage across the capacitor is fed to a full-wave rectifier. Its direct current output is connected across a regulator resistor and through a blocking rectifier to the exciter field. Because the polarity of the full-wave rectifier opposes the polarity of the exciter the rectifier voltage from the capacitor acts in opposition. Thus the net voltage on the exciter field is the difference between the opposing rectifier and exciter voltages.

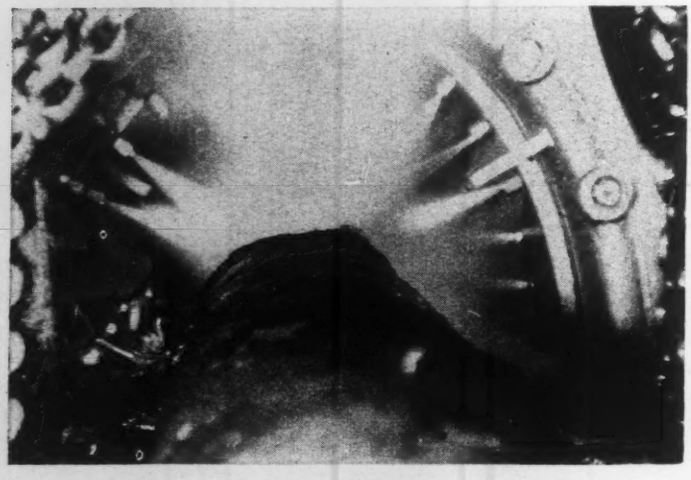
Change in generator voltage above or below normal due to change in load on the generator produces a corresponding amplified increase or decrease of voltage across the capacitor which, when impressed upon the rectifier, regulates the exciter. This rectifier voltage opposes the voltage applied to the exciter field winding, reducing the magnetic field flux of the exciter. This reduces the voltage across the field winding of the generator. In this way the generator voltage is held constant despite changes in load on the generator. The blocking rectifier acts as a valve to permit flow of current only in the proper direction, thus preventing injury to the full-wave rectifier should the exciter polarity become reversed momentarily.

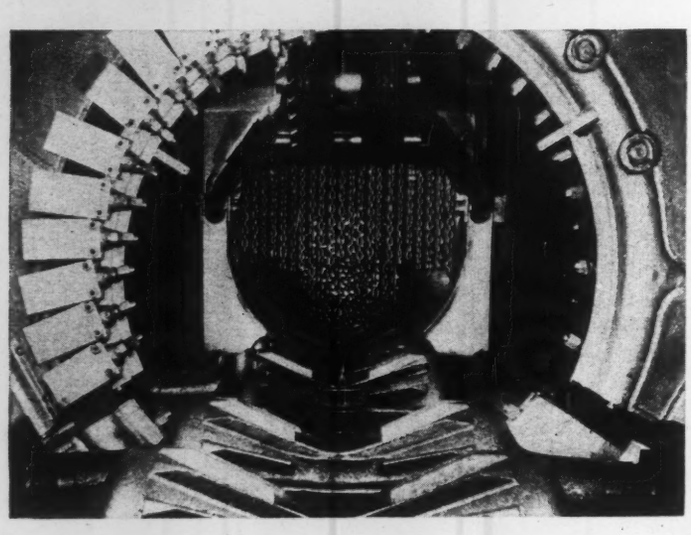




Plastic sleeves, which shrink to fit their forms at room temperature (left) provide insulation on connectors, cable ends and other parts of electrical equipment as well as "strip coatings" for protecting tube ends, precision machined parts, etc. These specially formulated vinyl plastic sleeves, developed by General Electric's plastics division, are swelled

in a dialator solution to 150 per cent of their normal diameter. When placed over a part, they shrink and take the permanent shape of the part as illustrated in the photographs. The plastic is tough, even when in the swollen state, and no splitting occurs during application. Tests indicate that the plastic will retain its good electrical, physical and chemical properties for many years. Unaffected by temperature and humidity the sleeves are resilient as well as acid and alkali resistant.

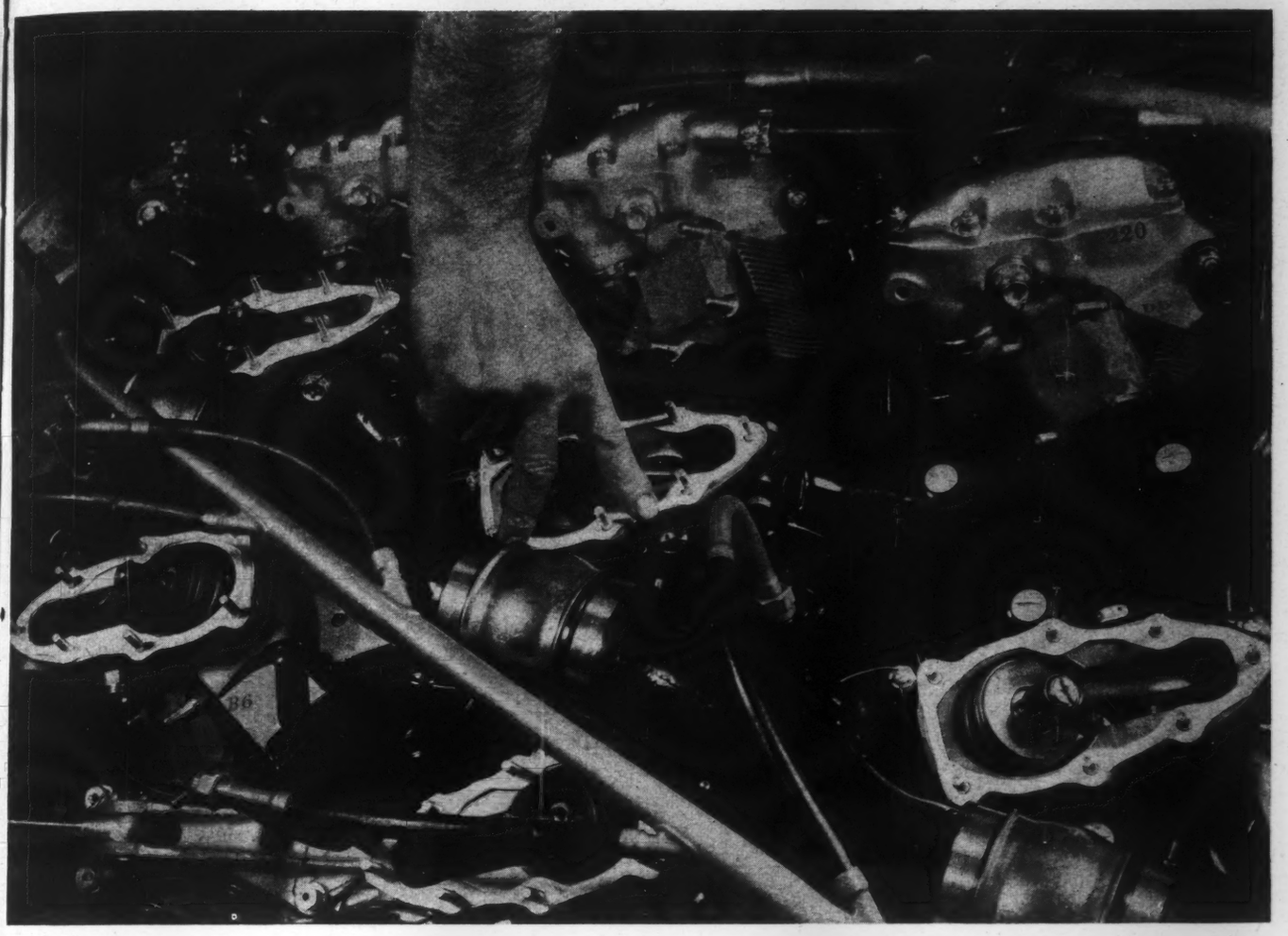




Jet barker above, peels logs up to 60 inches in diameter by means of water jets, at the rate of half-million board feet a day. Developed by the Worthington Pump & Machinery Corp. in collaboration with the Southview Pulp Co., this hydraulic machine consists of a ring formed by three 120-degree segments fitted with 36 nozzles. Logs, conveyed through the center of this ring at an average speed of 180 feet per minute, are stripped by impact of water jets exerting pressures of 1200 to 1400 psi with velocities over 400 feet per second.

Two of the jet segments are adjustable to accommodate various log diameters; also, number of nozzles may be selected according to log size. Two levers and pushbutton station, operated by one man, control the machine functions for delivering logs to and from the barker, sawing logs into convenient lengths, and gaging log diameter for nozzle adjustments. Two pumps are utilized, each being approximately 1000 horsepower and capable of pumping 1500 gallons per minute. The process has resulted in a 20 per cent saving of wood, compared with the usual process of cutting slabs to remove bark.

Built-in components that are so designed and mounted to facilitate maintenance and servicing gain customer goodwill and do much toward eliminating the aversion in many industries toward built-in equipment. Typical of the thought and planning that is going into modern designs is that reflected in the motor mounting shown below for a new Cincinnati No. 2 milling machine. By removing four bolts from the machine column, the motor and its mounting can be removed as one unit. This construction also serves as a convenient method of adjusting the tension of the V-belts.



SILICONE RUBBERS

. . . new gasketing materials for high-temperature uses

By John A. Moffitt
and Ahmed Panagrossi

The Connecticut Hard Rubber Co. New Haven, Conn.

PRIOR to the development of silicone rubbers, few materials were available to the designer to meet the critical demands presented by many high-temperature applications requiring resilient gasketing. Today, however, silicone rubbers are making possible improvements and refinements in the design of many machines by supplanting gaskets of other materials that were the limiting factor to further increases in operating temperatures.

Technically the silicones are poly-siloxanes. They are organo-silicon oxides polymerized in various chain lengths to give the qualities desired. Silicone rubbers are synthetic elastomers which

Fig. 1—Above—Installation of silicone-rubber-coated Fiberglas die-cut gaskets on the rocker box of Pratt & Whitney Wasp Major aircraft engine. Gaskets retain oil successfully under severe temperature conditions

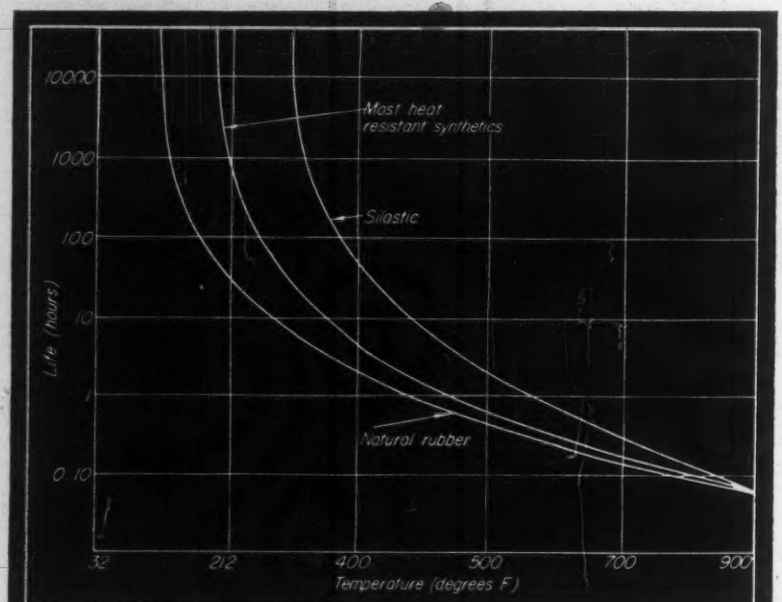


Fig. 2—Effect of high temperatures on the life of stocks of natural rubber, the most heat-resistant synthetic rubbers and Silastic

greatly resemble natural rubber and the other synthetic rubbers in that they are soft and flexible, resilient and rubbery, and can be stretched and compressed. However, because they are mainly of inorganic origin they have the following outstanding differences:

HEAT RESISTANCE: Natural rubber stocks usually become soft and sticky at about 225 F and hard and brittle at about −60 F. Special compounding of the regular synthetic rubbers has widened that range and it is possible to obtain stocks that will serve satisfactorily from about 275 F to −50 F. Silicone rubbers will remain resilient up to 500 F and flexible down to −70 F.

All the temperatures mentioned above should be considered as those present under usual operating conditions. Such temperatures are intermittent. For the silicone rubbers, for example, continuous, day in and day out, temperatures probably should not exceed 350 F. On the other hand short peaks considerably above 500 F can be withstood by the material although, of course, with a shortening of life, *Fig.* 2, depending upon the degree and length of time of exposure.

HIGH DIELECTRIC STRENGTH: Silicone rubbers have a high dielectric strength at room temperature and retain most of that strength at elevated temperatures. The fact that dielectrics made of other synthetic rubbers tend to fall off in resistance with increases in temperature is brought out in the comparison shown in *Fig.* 3. This quality of the silicones is of great importance in many of the new electrical applications.

Of equal importance in certain electrical applications is the fact that silicone rubber will not carbon track. A flash-over will merely burn out the organic molecules in the silicone and leave a chain of sand. The dielectric strength remains unimpaired.

MOISTURE RESISTANCE: Silicones are not only moisture resistant but they are actually water repellent. A film of moisture will not form over a clean surface to serve as an electrical conductor, nor will moisture seep through a silicone rubber gasket to reach the interior of a piece of equipment. Water absorption, as shown in TABLE I, is low for rubber-like substances. The material is also extremely resistant to ozone and ultraviolet rays, making it satisfactory for exposed uses.

CHEMICAL RESISTANCE: Silicone rubbers are not resistant to all chemicals and are affected by some that do not affect the other synthetic rubbers. But where they are resistant (as shown in TABLE II) they remain resistant at high temperatures. This permits the resilient gasketing of many pieces of chemical processing machinery.

ADHESION: High adhesion can be obtained between the silicone rubbers and glass, ceramics, iron and steel during the fabricating process. This property of adhesion is useful in working out special designs of gaskets and it permits reinforcement and protection from abrasion. It is difficult to obtain adhesion with the cured material.

NONVOLATILE: The low volatility of the silicones permits their use in vacuum apparatus without destructive effects of pre-extration.

TABLE I

Physical and Electrical Properties of Silastic†

Property	Type of Silastic and Stock No.							
	Coating & Laminating			Molding & Extruding				
	120	121	125	150	160	167	180	181
Density (at 25 C.)	1.7	2.3	1.7	1.9	2.1	2.0	1.6	1.3
Hardness (Shore A Scale)	50-60	85	40-50	45-55	55-65	55-65	75-85	60-70
Tensile Strength (psi)	500	600	400	400	600	600	700	475
Elongation (%)	150	50	200	300	200	110	75	90
Impact Resilience* (%)	45	—	52	53	52	54	47	—
Flex (DeMattia, Method B) % of Elongation No. of Cycles	50 500,000	— —	50 500,000	70 250,000	50 150,000	20 80,000	20 7,000	—
Water Absorption§ (%)	1.4	1.0	2.8	2.6	0.9	1.3	0.5	1.1
Water Absorption (m/sq. in.)‡	20	21	39	47	17	25	7	15
Dielectric Constant 10^6 cycles	7.5	—	7.5	5.6	9.0	9.8	4.6	3.0
Power Factor (%) 10^3 cycles	0.6	0.9	0.6	1.3	0.8	0.4	0.6	—
Power Factor (%) 10^6 cycles	0.1	—	0.1	1.0	2.0	0.1	0.3	0.3
Dielectric Strength (volts per mil)	1100	—	1100	500	500	500	650	—

†Silicone rubber made by Dow Corning Corp. *Bashore Resiliometer. §After 7 days immersion at room temperature.
‡Milligrams per square inch of exposed surface of ⅛-inch thick sheet sample after seven days in water at room temperature.

PHYSICAL CHARACTERISTICS: Silicone rubbers also have some physical properties that are not as satisfactory as those of natural or other synthetic rubbers. Tensile strength, although considerably improved in the newer compounds, is still below that obtainable with other rubbers. Also, their tear resistance and abrasion resistance are lower. Where higher values for these properties are required, re-enforcement, generally of Fiberglas, often is designed into the piece.

Compression-set at room temperature generally is greater than that of other high grade gasketing materials. But at higher temperatures the comparison swings in favor of the silicone rubbers. Above 200 F certain silicone compounds show considerably better resistance to permanent compression-set over extended periods of time than practically any other rubber-like gasketing material. The set characteristics of a silicone rubber compound are shown in *Fig.* 4.

Silastic—the silicone rubber made by Dow Corning corporation and the only one available in commercial quantities at the present time—is obtainable in eight different compounds, TABLE I. As shown, three of these compounds are for coating and laminating and are used by consumers mainly in the form of Silastic-coated Fiberglas sheets or tapes, which may also be die cut into gasket shapes.

The other five compounds may be used for molded gaskets, may be re-enforced where additional strength is required, or extruded in various shapes where gasketing of that type is indicated. Samples of Silastic gaskets made by these different processes are shown in *Figs.* 5 and 6. The range of physical properties of these eight compounds as shown in TABLE I is sufficiently broad to give the designer a choice of the type best for his requirements.

Current Applications Cited

Possibilities of improvement in the design of equipment through the use of gaskets with these properties are being studied in a large number of enginering departments. Silicone rubbers are new and it is probable that the most outstanding uses are still to be uncovered. But enough actual applications have been in production for upwards of a year to indicate the usefulness of the material in operating equipment. Among these applications are:

Molded Silastic Gasket For X-ray Equipment: This is a flat gasket molded to economize material. It is about 8 inches outside diameter and 5 inches inside diameter, and acts as an insulator for the housing. It is an excellent example of the use of silicone rubber as a high dielectric under elevated-temperature conditions.

Rockerbox Gasket on One of the Largest Aircraft Engines: This gasket, *Figs.* 1 and 5, which is die cut from a Silastic-coated fiberglas sheet serves to prevent oil leakage from the rockerbox under the temperature conditions that exist in a 3000-hp airplane installa-

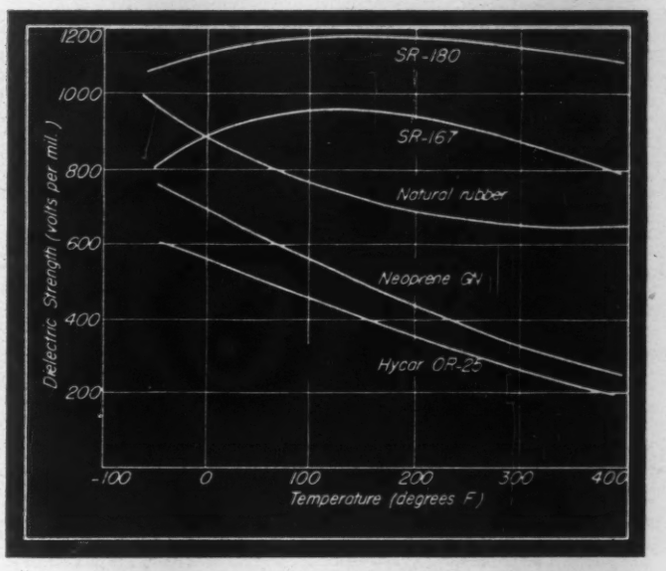


Fig. 3—Above—Change in dielectric strength, measured in peak volts, through changes in temperature on stocks made of natural rubber, Neoprene, Hycar and Silastic (silicone rubber) grades SR-180 and SR-167

TABLE II—Chemical Resistance of Silastic

Chemical Tested	Concentration (%)	Type of Silastic and Stock No.							
		Coating and Laminating			Molding and Extruding				
		120	121	125	150	160	167	180	181
Water		G	G	G	G	G	G	G	G
Hydrogen Peroxide	3	G	G	G	G	G	G	G	G
Ammonium Hydroxide	10	G	G	G	G	G	G	G	G
Ammonium Hydroxide	Conc.	F	G	F	G	G	G	G	G
Sodium Hydroxide	10	P	F	P	F	P	P	G	G
Sodium Hydroxide	50	G	G	G	G	G	G	G	G
Sodium Carbonate	2	G	G	G	G	G	G	G	G
Sodium Chloride	10	G	G	G	G	G	G	G	G
Sulfuric Acid	10	P	G	P	P	P	P	G	G
Sulfuric Acid	30	P	G	P	P	P	P	G	G
Sulfuric Acid	Conc.	P	P	P	P	P	P	G	P
Nitric Acid	10	F	G	F	G	G	P	G	G
Nitric Acid	Conc.	P	P	P	P	P	P	F	G
Hydrochloric Acid	10	F	G	F	G	F	P	G	G
Hydrochloric Acid	Conc.	P	P	P	P	P	P	F	G
Acetic Acid	5	G	G	G	P	F	G	G	G
Acetic Acid	Conc.	G	G	G	P	P	G	G	G
Toluene		P	P	P	P	P	P	P	P
Acetone		G	G	G	G	F	F	G	G
Gasoline		P	P	P	P	P	P	P	P
Stoddard Solvent		P	P	P	P	P	P	P	P
Mineral Oil		G	G	G	G	G	G	G	G
Carbon Tetrachloride		P	P	P	P	P	P	P	P
Sodium Hypochlorite	3	—	—	—	—	—	—	—	G
Sodium Hypochlorite	15	—	—	—	—	—	—	—	G
Formic Acid	5	—	—	—	—	—	—	—	G
Formic Acid	87	—	—	—	—	—	—	—	G
Tricresyl Phosphate		—	G	—	—	—	—	—	G

G—good: Test samples show a change of not more than 10% in weight or volume and retain at least two-thirds of their original Shore Efficiency.
F—fair: Test samples show a change of not more than 20% in weight or volume and retain at least one-third of their original Shore Efficiency.
P—poor: Above standards not met.

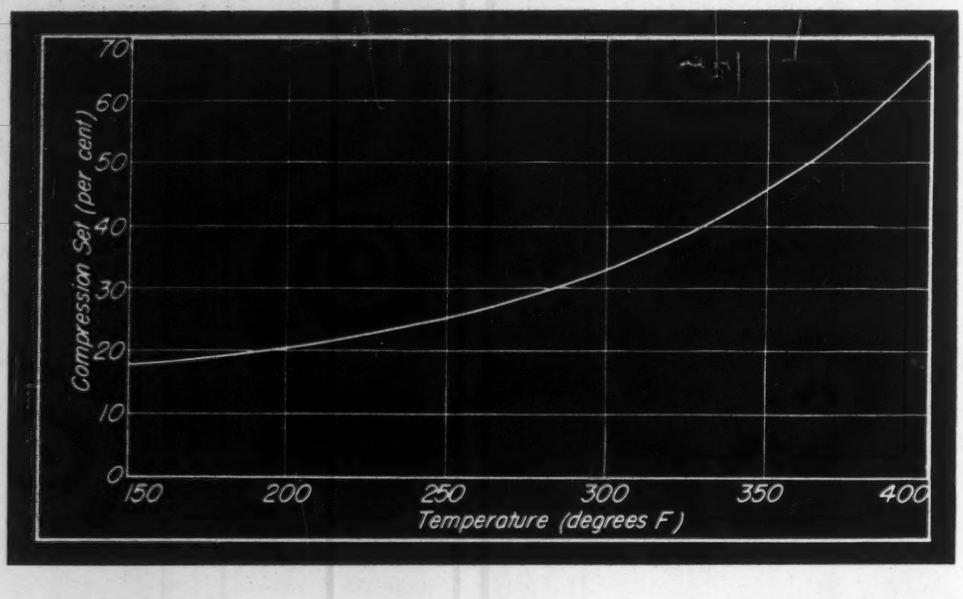
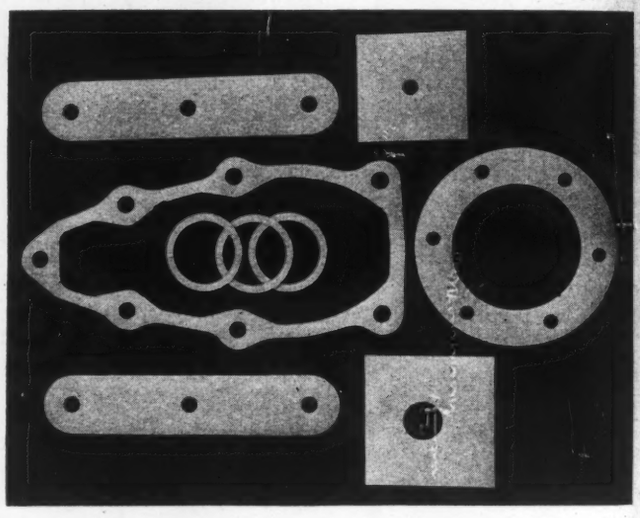


Fig. 4—Left—Increase in per cent of compression set of Silastic 181 measured according to ASTM D395-40T, Method B, at temperatures ranging from 150 to 400 F

Fig. 5—Below—Types of die-cut Silastic gaskets including aircraft engine rockerbox gasket of Fig. 1 at center



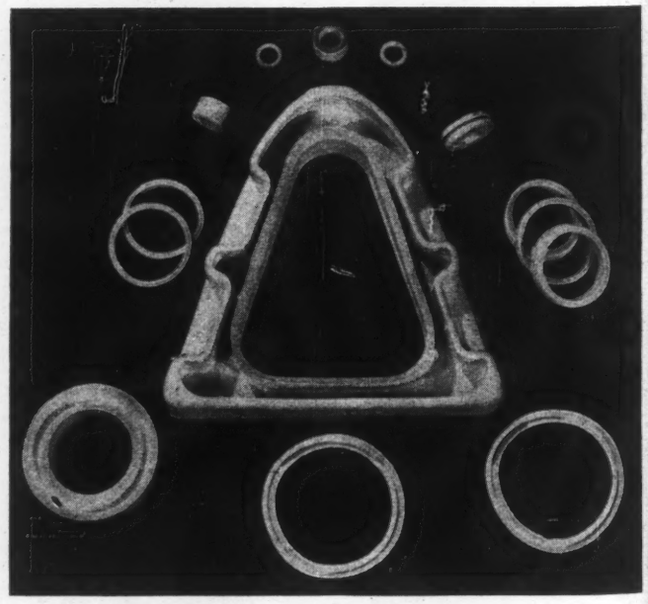


Fig. 6—Above—Variety of molded Silastic gaskets including large aircraft fin seal (center), pushrod packing, oil drain seals, V packings and others

tion. Service conditions appear to have no effect upon the ability of the gasket to retain oil.

Pushrod Packing and Oil-Drain Seal: Success of the rockerbox gasket led to the specification of silicone rubber in other parts of the same engine. The pushrod packing, *Fig.* 6, is a molded ring designed into a light metal stamping to overcome the low abrasion resistance of the material. The oil-drain seal, *Fig.* 6, also acts as a spacer and is molded with high adhesion to two metal end pieces. Because the Silastic could withstand the oil temperatures and still give a perfect seal it permitted a simplification and improvement in the design of the drain assembly.

Steam Generator Packing: In this unit the electrodes pass through the boiler shell and generate pressure steam. The Silastic packing insulates the electrodes from the shell and seals against the steam pressure. This is a good illustration of the use of a combination of the properties of Silastic. It is doubtful if any other material has the same insulating, temperature resisting and sealing qualities necessary to make this unit operate successfully.

Leading Edge Fin Seal: The rather intricate molded seal shown in *Fig.* 6 is used in the folding wings of a carrier plane. One seal is placed on each side of the wing at the point of folding and they seal, in the closed position, the heater duct for the de-icer.

Gas Cylinder Gasket: Extremely low temperatures are created when pressures of bottled gases are released. There was need for a gasket that would not get hard and brittle at these temperatures. The silicone rubbers, remaining soft and flexible down to −70 F, successfully solved this problem.

Oil Burner Seal: A small, space saving, oil burning hot air heater required a gasket that would seal against fuel oil and withstand the high surrounding temperature of this compact piece of equipment. Silicone rubber gaskets permitted the design to be worked out within the small size desired. The unit has passed the Board of Underwriters requirements and is now on the market.

As these examples indicate, Silastic is specified mainly where no other rubber will serve. Its high cost prevents its use merely as a substitute for the usual synthetics. It seems probable that in the future this high cost may be somewhat reduced through larger production of the raw

material and through processing short cuts that will be developed in time. But at present, parts of Silastic must carry a cost of approximately ten times that of similar parts of high grade synthetics.

In many installations it is possible to gain the better properties of Silastic and still keep costs in line by using smaller or thinner gaskets. Designers working with ordinary rubber gaskets, where costs are less, frequently specify thicker materials than really are needed to assure a perfect seal. With Silastic these savings become important and the "spring" of a properly cured sheet of silicone rubber is such that a thin gasket usually will do the job.

Correct re-enforcement of Silastic is also important particularly in the smaller cross sections. Fiberglas is the principal re-enforcement material, mainly because of its excellent heat resistance, but other materials frequently are used as re-enforcements where special properties are desired. Silastic on Nylon makes a very flexible sheet that bends with small pressures and has long flexlife. Silastic-coated woven metal fabrics have extremely high tensiles, making them of interest as a pressure gasketing material. Re-enforcement with "A" fiber is being used experimentally and offers possibilities as an extremely flexible gasket with high temperature resistance.

Care should be taken in the design of the more complicated molded gaskets as re-enforcement is difficult and sometimes impossible. Even without re-enforcement it is well to avoid undercuts or deeply molded sections. Where these are required allow plenty of taper since the silicones do not mold as easily as rubber and are much more tender in the uncured state.

Because the silicones are new, full knowledge of their properties has not been developed. There is not sufficient experience to state definitely that gaskets will or will not work satisfactorily in many of the new applications that are suggested. A test, preferably on the specific piece of equipment under operating conditions, often is the only way to obtain the proper answer.

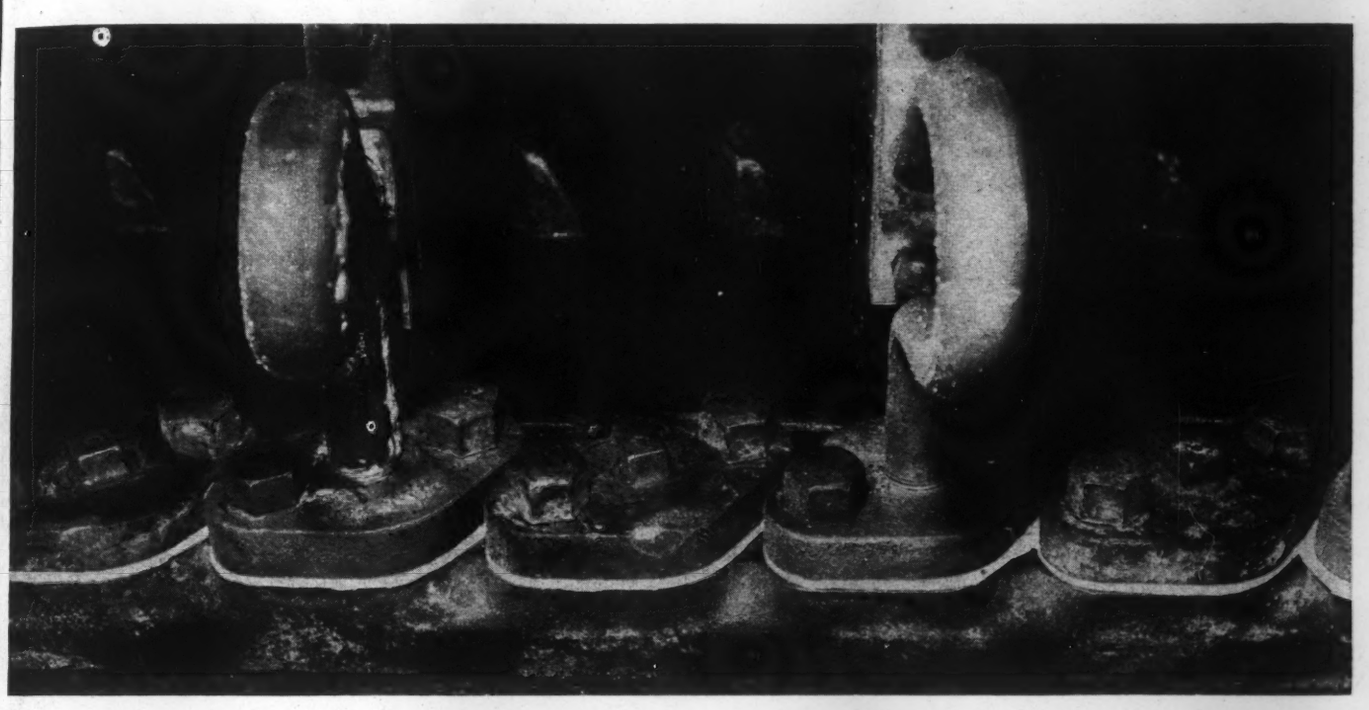


Fig. 7—Gaskets of Silastic 181 used to seal steam-heated vacuum oven. Both high temperatures and a considerable pressure differential complicate his application in which any leakage of steam would spoil the vacuum

Rotary-Wing Kite Being Studied at Wright Field

UNCONVENTIONAL type aircraft, resembling a combination kite and helicopter, which was used by the Germans in submarine air observation, is being studied and tested by Technical Intelligence, Air Materiel Command, Wright Field, O.

This strange craft, dubbed the German Submarine Rotary Kite by Intelligence, and called "Sandpiper" by the Germans, weighs only 180 pounds and has no motor. It consists of a pilot's seat, pylon, conventional autogiro head rotor, a horizontal supporting tube extending aft, and a rudder. Controls are the conventional stick and rudder pedals and the landing gear consists of skids which can be folded parallelogram fashion.

To initiate and maintain flight, the kite is towed by a submarine. Air pressure on the three-bladed rotor will create approximately 205 rpm with an air speed of 25 mph. The entire craft can be knocked down to a compact package and stored in a submarine. Helicopter pilots find it easy to fly, and it is believed several crew members on each submarine were trained to operate the craft.

By Mark Morgan
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Electronic Motor

. . . no longer a novelty, provides unusually simple and flexible feed drive for new vertical milling machine

SINCE the time of its inception, electronic motor control has been favorably accepted by machine tool builders and users, its advantages over older types of drives having been immediately recognized. This was in spite of the fact that electronic equipment in general has control apparatus and a wiring diagram which, at first, seem rather complicated when compared to more conventional schemes.

A new small vertical miller which incorporates all the requirements of a modern machine and all the features which go with an electronic variable-feed drive has recently been introduced by the Reed-Prentice Corp. The wiring diagram is comparatively simple and the amount of control equipment, *Fig.* 1, is moderate both in quantity and volume, with no need for crowding into inadequate space. Representing a simple and relatively inexpensive application of electronics, the control features of this machine should be of considerable interest to designers of other types of machines.

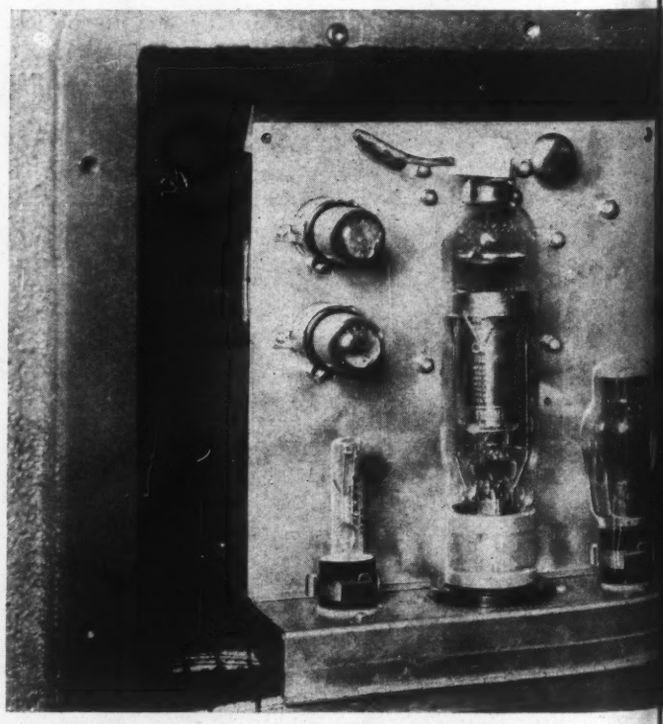
MACHINE SPECIFICATIONS AND FEATURES: This tool room miller, shown in *Fig.* 2, will do face and straddle milling, as well as boring, drilling, die sinking and duplicating. A solid table base replaces the conventional overhung knee, and houses the feed motor. A quill-type milling and boring head is mounted on a sliding ram which is power driven up and down by means of a lead screw; it is adjustable horizontally by hand. The head is equipped with a tilting arrangement to shift the spindle axis 45 degrees in the front-to-back plane either side of vertical, and 30 degrees in the left-to-right plane, also either side of vertical, *Figs.* 2 and 5.

The spindle is powered by a 3-hp, 1800-rpm induction motor, with spindle speeds variable between 80 and 2300 rpm. This is accomplished through a Reeves drive, stepless adjustments being made by two jogging pushbuttons marked "faster" and "slower".

Another motor serves for power feed and rapid traverse and can be applied to the table drive for either longitudinal or cross feed, or to the ram for up and down feed. Selection of the desired motion is accomplished by shifting the two control knobs at the front of the machine, *Fig.* 2.

For machining accuracy, especially when doing die sinking work with an automatic tracer, it is essential that all chatter or vibration be eliminated. Inasmuch as this can best be accomplished by small adjustments of feed rate, or of spindle speed, or both, the best and fastest results are reached when the machine operator has means to make such adjustments while milling, without having to stop the

Fig. 1—Below—Electronic control unit mounted in a 15½-in. wide by 20-in. high cavity in column of vertical milling machine. Thyratron C6J in center is armature rectifier tube; smaller tube 5Z3 at right is field rec ifier. Unit also inc udes thermo-relay for filament warm-up, lower left, and two auxiliary control tubes, upper left



Control

machine. For this reason it is considered important to have a centralized pushbutton control station, as shown in *Fig.* 2, within easy reach of the operator. All buttons for spindle and feed control, as well as the large potentiometer knob for selection of feed rate, are located on this panel.

Directional control of feed is accomplished through a five-position lever also mounted in front of the machine, *Fig.* 2. By means of cams the lever operates three limit switches

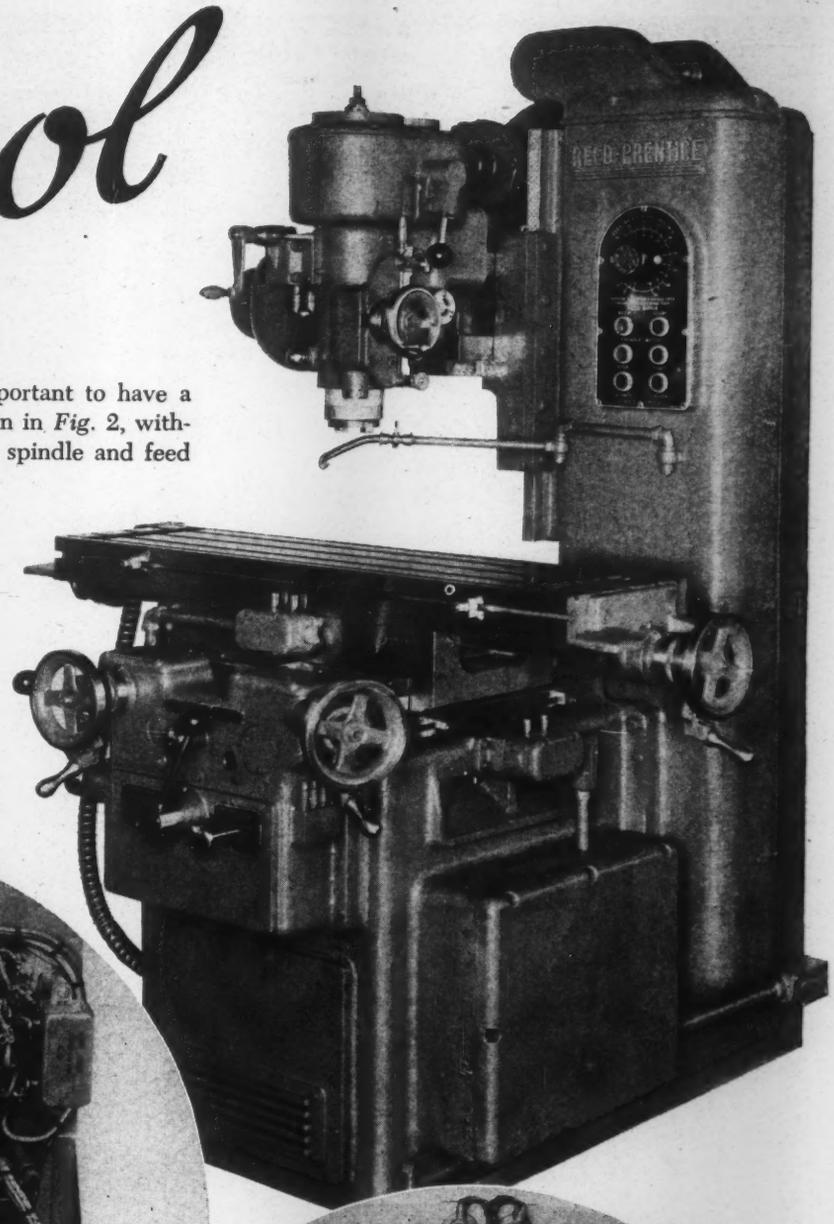


Fig. 2—Right—Reed-Prentice No. 12V vertical milling machine, which employs electronic control for feed drive

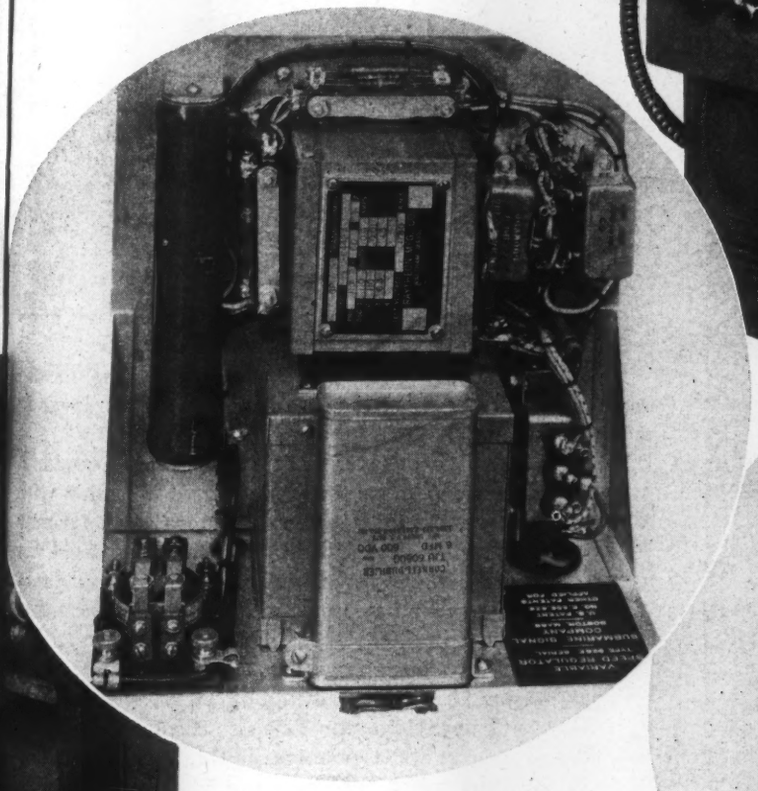


Fig. 3—Above—Back view of electronic unit, which is mounted on a hinged panel

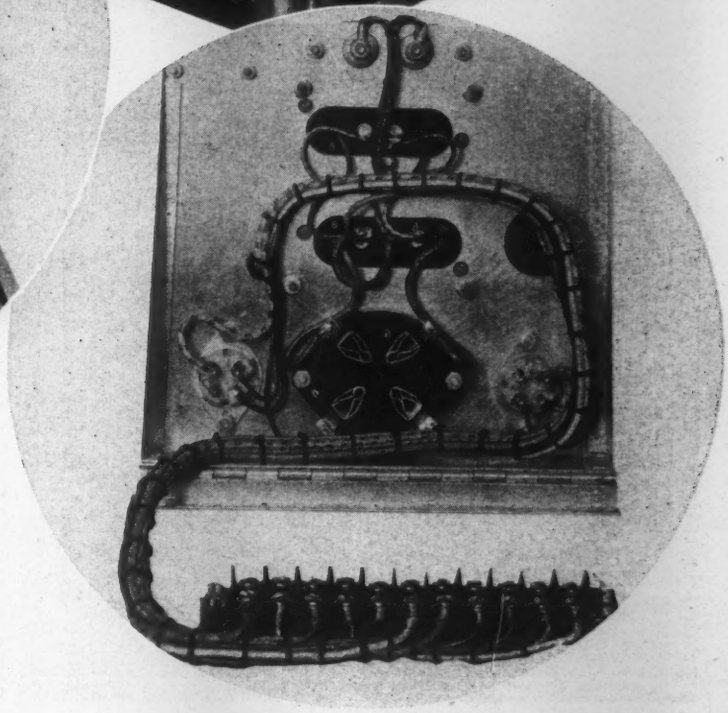
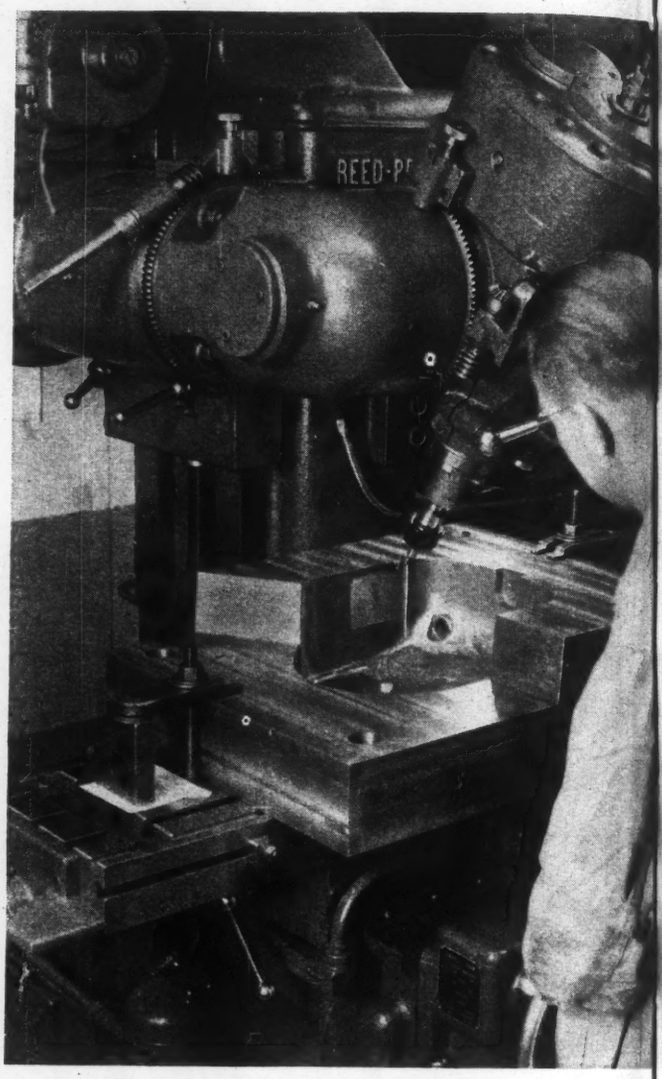


Fig. 4—Right—Bottom view of electronic unit. Photos, Figs. 3 and 4, courtesy Submarine Signal Co.

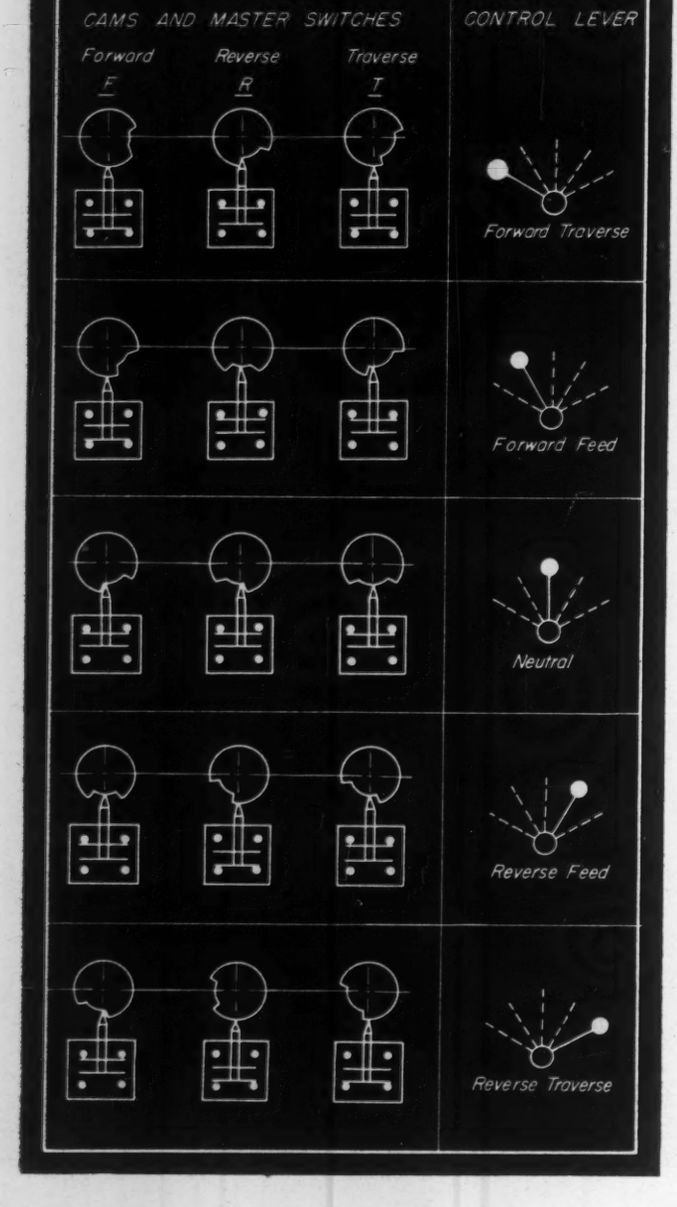
Fig. 5—Right—Typical die-sinking job on 12 V mi ler, using the tilting arrangement of the spindle head



F (forward), *R* (reverse), and *T* (traverse), *Figs.* 6 and 7. In the first position of the lever, "Forward Feed" (or "Reverse Feed") either side of neutral limit switch *F* (or *R*) is tripped, contactor *F* (or *R*) energized and motor armature circuit closed. The next position of the lever (further out of neutral) is for traversing, and at this time both *F* and *T* (or *R* and *T*) are energized. Returning the lever to neutral opens contacts *F* or *R* in motor armature circuit and closes dynamic brake resistance across the armature.

Selector switch marked "Power Feed", *Fig.* 7, energizes contactor *M* which in turn closes the filaments circuit and initiates the warm up of the tubes; it also energizes the motor field circuit. It will be observed that *M* is energized through interlocking contacts, 18-20, which are open whenever the operating levers are not in neutral. This is a safety measure to prevent a runaway machine in the event the operator should inadvertently forget the lever in a feed position.

Fig. 6—Below—Master control switches and control lever sequence for feed or traverse in either direction



Three pairs of overtravel limit switches are provided, with adjustable trip dogs; they are wired in series with contact *F* (or *R*), as shown in the diagram, *Fig.* 7. It may be observed that no electrical interlock has been used for contactors *F* and *R*, contrary to what is conventional practice, because they are operated by a single lever, in either the left or right position; this, of course, rules out any chance of simultaneous energization.

The "Master Stop" button shuts off all circuits, including the tube filaments circuit. This also is contrary to what is probably conventional practice, but it was made possible because the tubes employed on this machine require a short warm-up time, as will be discussed later.

ELECTRONIC FEED DRIVE: The feed motor is rated 1 hp, 1800 rpm, 230-volt dc, and has class B insulation. The electronic unit is a single-phase, half-wave type. Reed-Prentice has also used, on other machines, single-phase full-wave drives, while three-phase half-wave units are also available and popular. The proper selection depends, of course, on the type of machine and the work to be performed.

It is known that tube current rectification, if unfiltered, results in a ripple whose frequency depends upon the number of rectifying tubes, and is equal to 60 cycles per second in half-wave single-phase systems, 120 cps in full-wave single-phase, and 180 cps in three-phase half-wave systems. It is also known that the lower the frequency of the ripple, the greater its amplitude, and vice versa. This ripple, under certain conditions, may induce a pulsation or vibration of equivalent frequency into the cutting tool or into the work, which may in turn result in poorly machined surfaces.

Experience on a large number of Reed-Prentice toolroom milling machines has shown that a half-wave single-phase job is perfectly adequate for the kind of work performed by such machines. All other considerations aside, this allows for a considerable saving in the space required by the electronic unit, and is important in the application to small machines such as the No. 12 Miller. *Fig.* 1 shows the electronic unit mounted in the column, readily accessible and serviceable through a hinged panel construction. A study made to ascertain whether a full-wave unit might have been located in the same space showed that it would have been impossible to do so.

The feed drive motor of a milling machine, in addition to accelerating the moving masses, is also called upon to overcome the reaction of the cutter against the work plus the mechanical attritions of the drive. Generally speaking, the motor shaft torque due to the reaction of the cutter is only a fraction of that caused by the attritions of the drive; assuming that friction does not vary with speed, it is therefore evident that in a feed drive the torque is virtually constant while the horsepower increases with the speed, *Fig.* 8. This condition is confirmed by the test results shown in TABLE I. Values tabulated in the armature current column are proportional to the torque developed by the motor, because the field excitation is held constant through half-wave rectifying tube 5Z3; and it will be noted that, for all practical purposes, the torque does not vary with the varying of the speed.

One of the outstanding features of this drive is brought out by these test data,

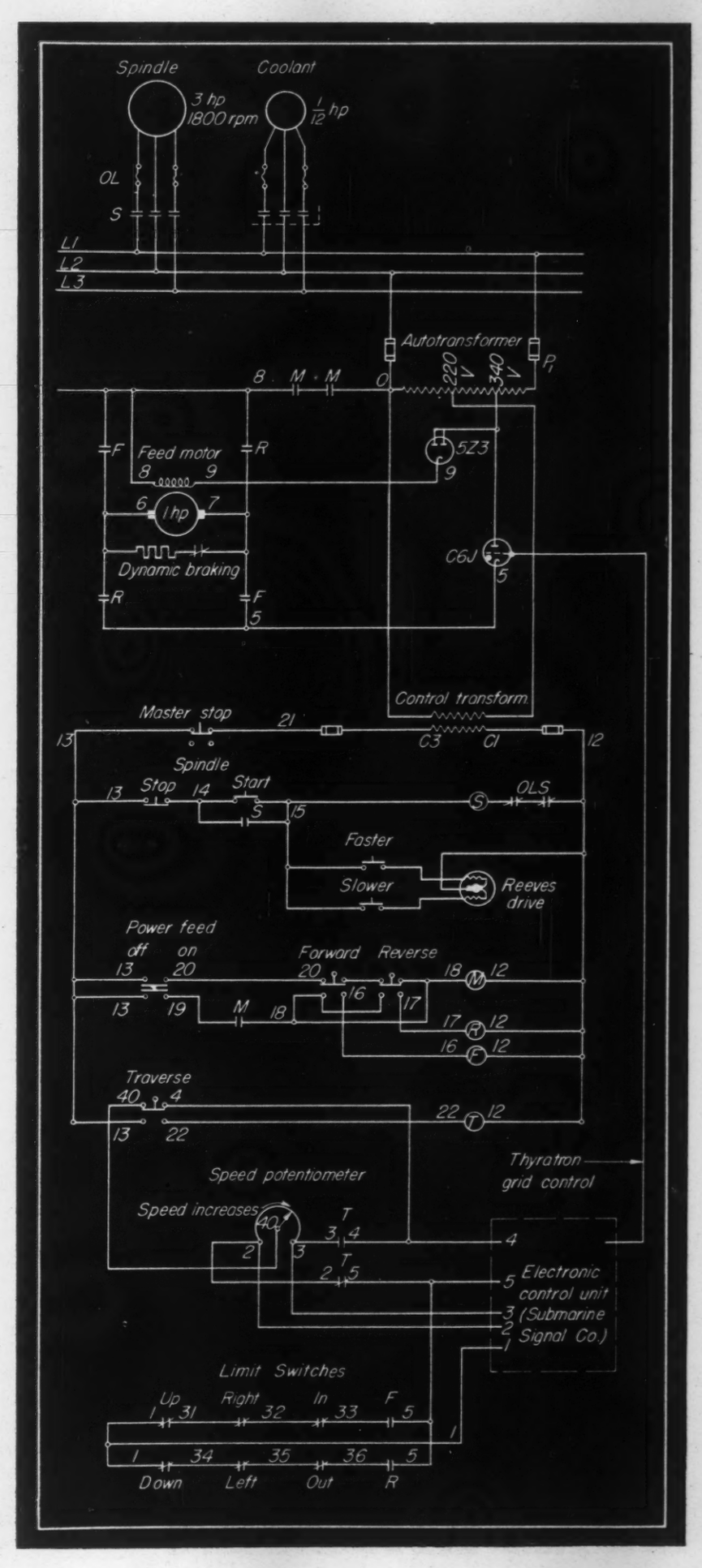


Fig. 7—Right—Elementary wiring diagram, showing the control circuits for the spindle and feed motors

namely, the wide speed range obtainable at constant torque, with armature voltage control only. It appears that the variable speed feed range is almost equal to 100:1 while the maximum (traverse) to minimum speed ratio is around 130:1. The high value of this ratio, together with the fact that the motor can develop the maximum torque even at top speeds, made it possible to employ the same motor for both feeding and traversing. To accomplish the two different functions only the addition of a relay, a limit switch and a resistor was necessary. From the point of view of simplification, this is a consideration not to be overlooked.

Effect of Size on Traverse Speed

It might be argued that a traverse speed of 33 inches per minute is still rather low as milling machine practice goes; however, considering that the range of the table is only 28 inches longitudinally, and 12 inches crosswise, this speed is deemed adequate. Larger machines, of course, require much greater rapid traverse speeds. To obtain these, mechanical means such as back gears can be used, or electrical means such as motor field control, or electro-mechanical means such as the addition of a clutch and a separate a-c induction motor with its associated magnetic equipment.

Extended discussion of these and other rapid traverse schemes would be out of place in this article. But it may be noted that, up to this time, field control can provide a maximum speed range of only 4:1 which may be still rather low for some applications. Lately, however, new schemes for motor field construction and interconnection have been proposed, whereby a field weakening range up to 8:1 is obtainable. Progress along these lines is being watched with great interest by designers in the machine tool field.

An important consideration should be borne in mind when applying the weak field range of the motor to a machine feed drive, namely, the fact that torque falls off approximately with the field current. Thus if the designer chooses to make wide use of field control in order to reach high speeds, it may be necessary to employ oversized motors, because the torque at maximum field rather than the torque at minimum field is what determines the frame size of a motor.

This condition is represented graphically in *Fig.* 9 from which it may be seen that motor No. 1, having frame size determined by torque *OA*, is adequate for a maximum speed *OC*. When the speed is increased by an amount *CH* through field weakening, the torque of motor No. 1 would fall under the required value by an amount represented by *DL*. It is then necessary to employ motor No. 2, with a larger frame size, determined by torque *OE*. But in this event motor No. 2 would be underloaded throughout its operating range, and the potential horsepower represented by area *AEFGA* remains unused.

Basic Electronic Circuit Is Simple

ELECTRONIC UNIT AND TUBES: A detailed explanation of the electronic diagram, *Fig.* 7, would be uncalled for here, but it may be noted that the basic circuit is very simple, as the grid control for the thyratron is derived from the counter-emf of the motor.

A group of fixed resistances is connected across the motor armature as a voltage divider; its potential, of course, varies with motor counter-emf, which in turn is proportional to speed. The voltage signal of these resistances is connected to the thyratron grid. Another voltage signal determined by the position of the speed potentiometer is also connected to the thyratron grid, its action being in bucking relation to the one noted above. Now, if any extraneous cause tends to change the speed of the motor from the preset value called for by the potentiometer, the thyratron plate voltage is varied in the opposite direction, until equilibrium is re-established.

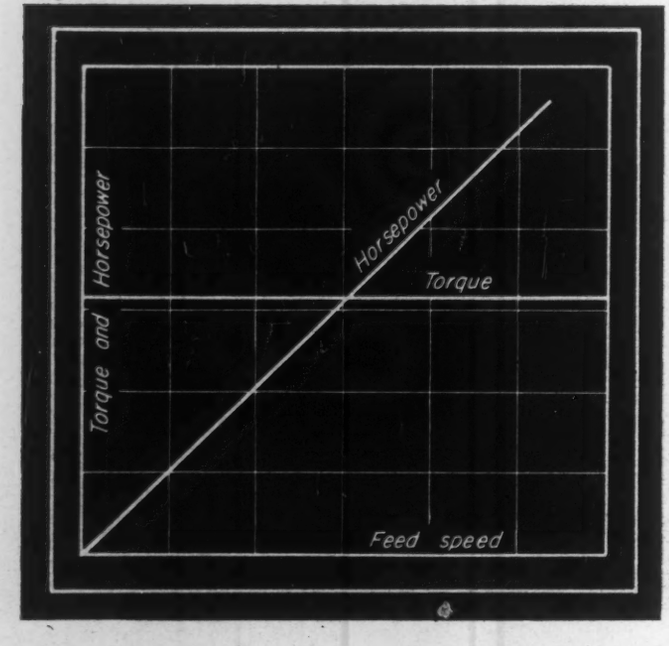
The electronic unit incorporates the following features which are a "must" in a feed drive: (*a*) Dynamic braking, to prevent coasting; (*b*) current-limiting circuit, for the protection of motor and thyratron, and for holding the motor armature current down to a safe value when the motor is started, stalled, or subjected to a sudden load; (*c*) a flat load-speed curve which means small speed variations from no-load to full-load condition.

Inspection of *Fig.* 1 shows the type of rectifying thyra-

TABLE I
Motor Performance with Armature Voltage Control

	Armature		Speed	Feed Rate (in. per min.)
	(volts)	(amperes)	(rpm)	
RAM UP				
Minimum feed speed	4	1.25	20	1/12
Maximum feed speed	240	1.25	1900	16
Traverse	350	1.35	2600	22
TABLE				
Minimum feed speed	4	.85	20	¼
Maximum feed speed	240	.85	1900	24
Traverse	350	1.00	2600	33

Fig 8—Horsepower and torque requirements of milling machine feed drives, torque being virtually constant



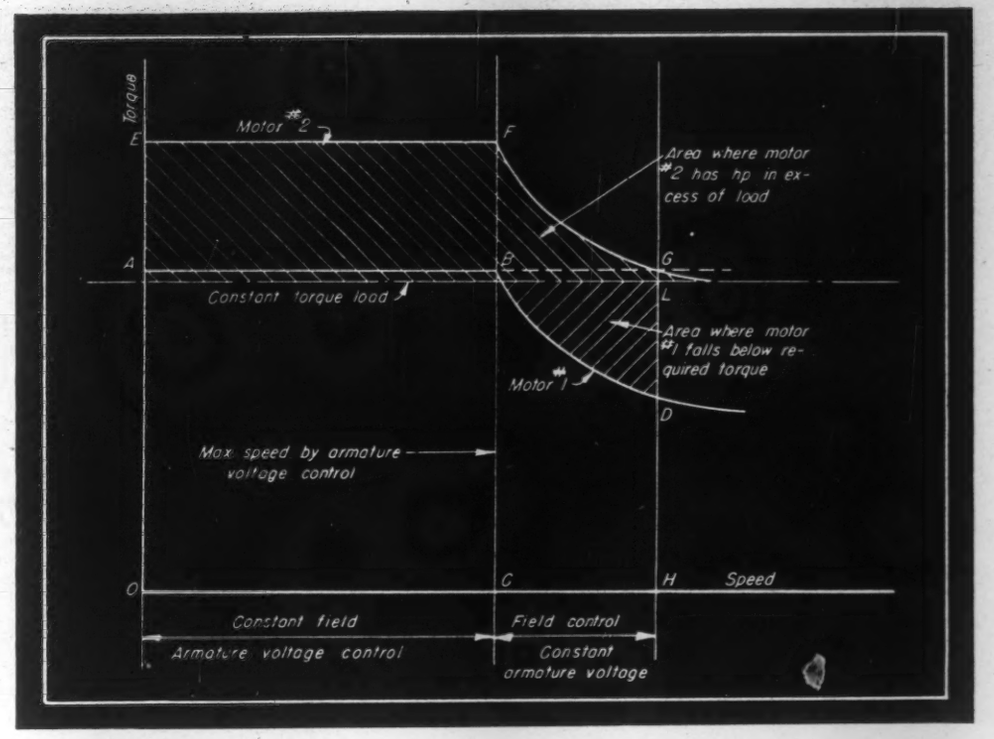


Fig. 9—Comparison of speed-torque curves for two different motors, showing torque reduction with field control

tron used in the motor armature circuit; this is a glass tube filled with Xenon gas. On similar applications, i.e., motor drives rated 1 or 1½ hp, another type of tube—a mercury vapor-filled thyratron in a metal envelope—is also quite commonly used.

Xenon and Mercury-Filled Tubes Compared

These two types of grid-controlled gas rectifier tubes have similar electrical characteristics, and in particular have identical value of maximum average plate current 6.4 amperes, dc, which makes them well suited for armature control of 1½-hp, 230-volt d-c motors. Their behavior in actual operation has been observed over a period of time long enough to draw the conclusion that they are both to be considered highly satisfactory for this type of application.

However, there are some basic differences which it may be worth while to note from the point of view of the machine designer. Contrary to what was feared when electronic motor control was first brought out, other devices, including coils and contacts of conventional magnetic relays, are likely to give trouble before a tube fails. However, the life of the power tubes still is an important factor. The mercury-vapor tube is likely to have a life several times longer than that of the gas-filled tube. On the other hand, the mercury vapor tube is approximately one-and-a-half times more expensive than the gas tube, and its outline dimensions are somewhat larger than those of the gas tube.

Cathode warm-up time of the mercury tube is about 5 minutes (this time being required to vaporize fully the mercury which has collected at the bottom of the tube while cold); a synchronous motor driven timer is usually employed to measure this interval. For the gas-filled thyratron, however, a warm-up time of only 45 seconds is adequate. To measure this interval a vacuum bimetallic thermo-relay, shown at bottom left side of *Fig.* 1, may be used. This means less auxiliary devices and a simplification in the diagram.

Although the question of tube life is, of course, of interest more from the point of view of machine upkeep than of machine design, the following can be pointed out: When a tube manufacturer gives a tube life guarantee of 2000 hours (such as in the case of the thyratron employed in the machine under discussion), this is considered as a minimum value, the probable life usually being much longer; the minimum life value is to be expected with the full rated load actually applied. In machine tool duty, the electronic unit may be energized and the filaments lit for periods of time when no power is delivered by the tubes; moreover, it is good design practice to employ a tube and a motor at less than maximum rated load. Both of these factors work in favor of longer tube life.

Underloading Promotes Long Tube Life

It may be observed that on the No. 12V Miller, the tube has a current rating of 6.4 amperes, while the 1-hp motor has a rating of approximately 4 amperes, the drive actually requiring a delivery of around ½ to ¾ hp. Under this condition both motor and tube are underloaded, and a long tube life is to be expected.

This is proved by the following observations: Reed-Prentice had two drives, of the same rating and type as used on this machine, installed in their own plant; one operated for two years, the other for approximately one year. The first machine was in operation at least 5000 hours and the second machine 2500 hours. During most of this time the electronic units were energized, i.e., the filaments warm, although it was to be assumed that only 25 per cent of the time the tubes were delivering plate current. During this period no thyratron failure was experienced, which led to the conclusion that, in such a case, the replacement of tubes is a matter of relatively small concern.

Electric Heaters and Their Control

How to determine capacity, type and location of heaters and thermostats

By J. W. Harrison
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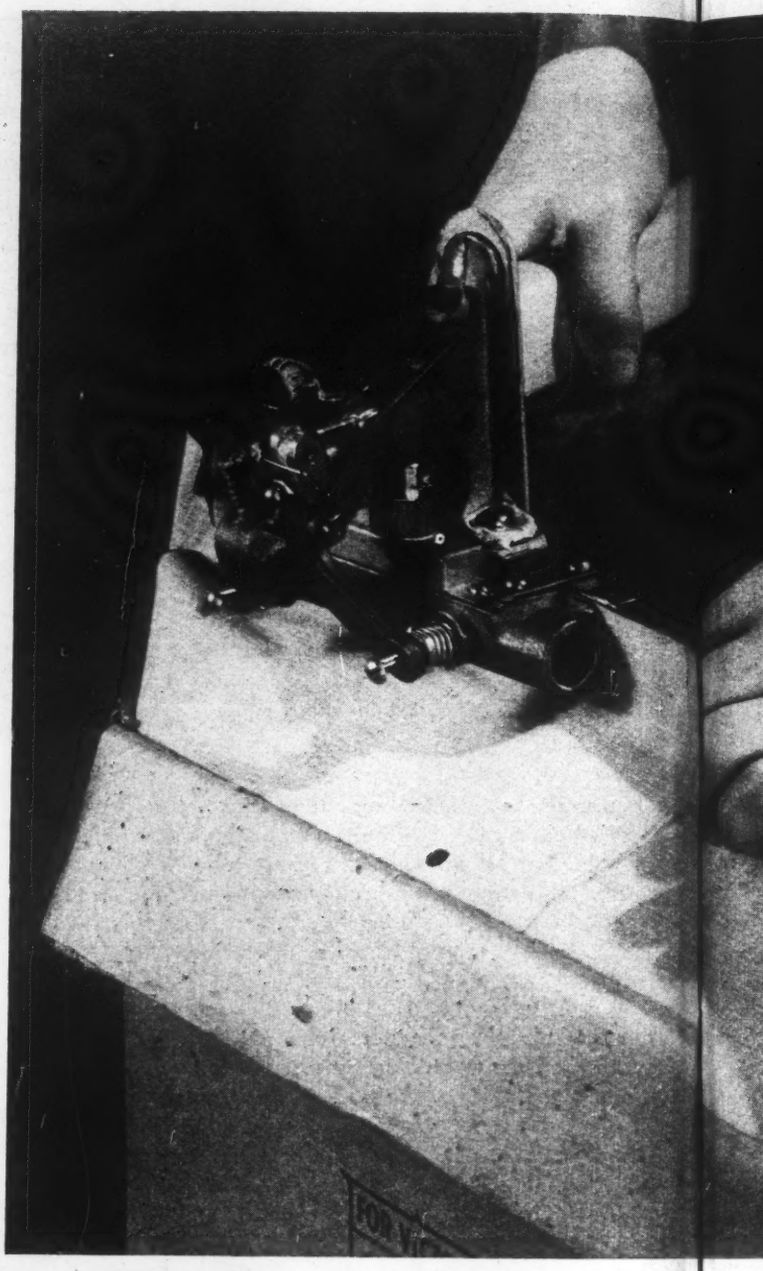


ACCURATE temperature control of electrically heated tools (see *Fig.* 1), tanks and hoppers is attained when the heating element and the bimetallic thermostat are designed to as-

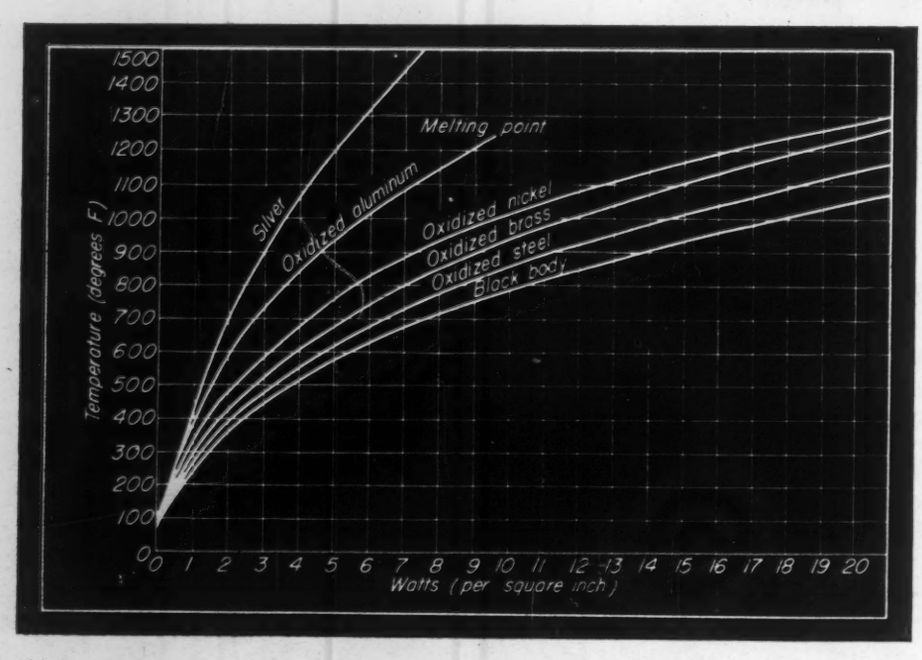


Fig. 1—Above—Package sealing machine heated by two cartridge heaters. Close temperature control is due partly to the fact that bimetal carries the current, thus acting as a booster heater

Fig. 2—Left—Heat losses from a flat or curved vertical surface are 100 per cent of curve value. For a horizontal surface they are 110 per cent of curve values for upper face and 55 per cent for lower

sure balance of the thermal relation between the two components. Factors which determine the type of heating element to be used are: (1) Size and shape of the part to be heated; (2) material from which the part is made; (3) operating temperature the part requires; and (4) wattage required to supply the heat necessary including radiation losses. Factors which determine the type of bimetallic thermostat to be used are: (1) Temperature limits; (2) current rating; (3) thermal capacity and mass of part and material to be heated; and (4) thermal contact necessary for accurate control.

Fig. 3—Right—Mica insulated element that heats stencil ring by radiation

Wattage, P, required to heat materials may be calculated by substituting in the following formula

$$P=\frac{wcT}{3.415}+R$$

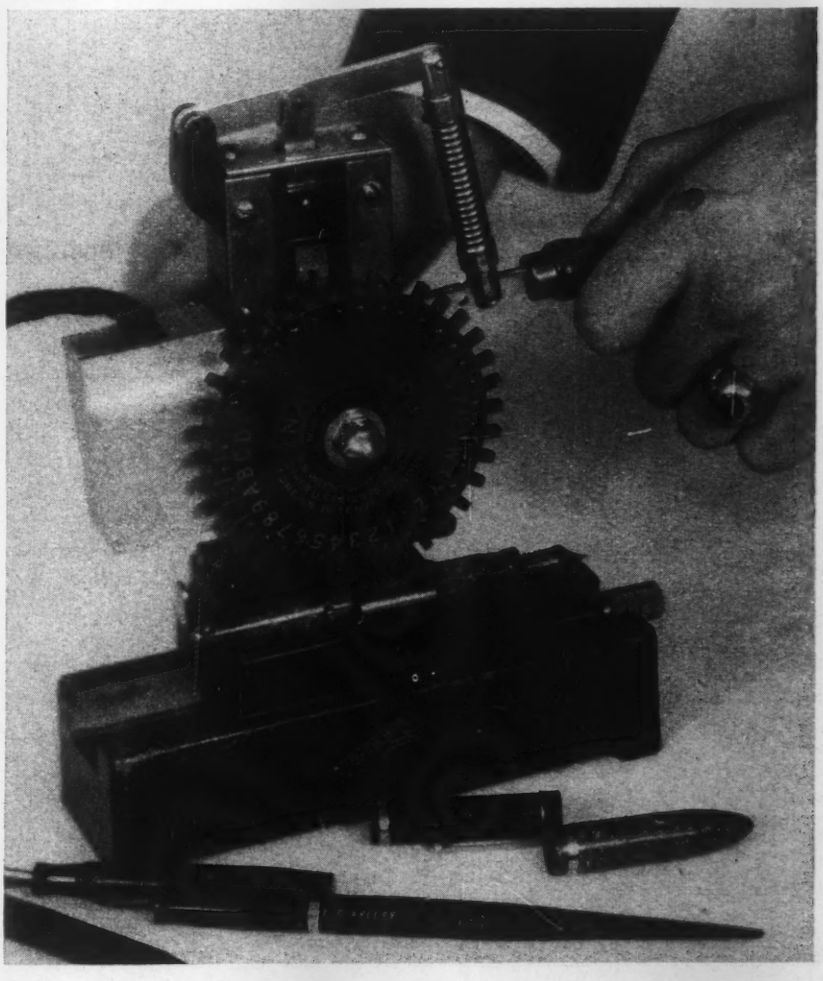
where w is weight of material (lb/hr), c is specific heat of material (Btu/lb/°F), T is temperature rise (degrees Fahr.) and R is losses from radiating surface (watts). When change of state of material occurs the power required for fusing or vaporizing must be added. Radiation losses may be estimated by calculating the square inches of radiating surface and multiplying by the watts loss per square inch for the proper curve in *Fig.* 2. The total wattage to install must, therefore, be equal to or greater than that required to heat the material and unit as well as supply the radiation losses.

When thermostatic temperature control is contemplated, the installed wattage must exceed the exact wattage required for processing because: (1) Theoretically the thermostat will not be called upon to function when material is being processed if the exact wattage required is all that is installed; (2) processing demands are such, due to inherent material characteristics that a safety factor of as much as 25 per cent excess wattage is desirable. The latter provision makes possible the use of standard heating elements, thereby assuring an economical installation.

The time required to heat a tool can be estimated readily inasmuch as this heating is done prior to operation and the wattage has been determined for heating the material being processed. Time, t, in hours becomes

$$t=\frac{1}{3.415}\left(\frac{Wc_1T}{P-\frac{1}{2}R}\right)$$

where W is weight of heated part of tool (lb), c_1 is specific heat of tool



(Btu/lb/°F), T is temperature rise (degrees Fahr.), P is the wattage applied, and R is the radiation loss calculated from *Fig.* 2.

Cartridge type heaters are ideal for heating die blocks or molds which are of sufficient size to permit their use as shown in the sealing machine in *Fig.* 1. These heaters are efficient for operating temperatures up to 750 F, maximum. Installation requires the drilling of a hole in the platen and then inserting the cartridge heater. The mounting hole should be drilled with a fractional size drill equal to the diameter of the heater to be used. Standard cartridge heaters are made undersize to assure the required snug fit and are made in sizes varying from ½ to 1 5/16-inch in diameter, 2½ to 10 or more inches in length. Wattage ratings vary from 75 to 1500 watts for 115 or 230-volt operation.

Utilizes Mica Insulated Heater

Mica insulated heaters as employed in the stencil machine shown in *Fig.* 3 are made in special sizes and shapes for temperatures up to 650 F to meet the requirements of the specific application. Special elements of this type can be made in small quantities with little or no tool expense involved. The heating element is sandwiched between two sheets of mica and is connected to the power source by appropriate terminals or insulated wire leads. This element may be nickel-chromium wire or ribbon wound on a mica form, the complete unit being held together with eyelets or by special cement applied to hold the mica together until it is assembled into the device to be heated. This type of heater must be clamped against a smooth surface and held tightly with a clamping plate. An asbestos pad ⅛-inch thick, between the element and clamping plate will help assure uniform contact pressure between the element and its mounting surface and will also serve as effective thermal insulation.

Strip Heaters Have Wide Application

Strip heaters find their greatest industrial application in the heating of large molds and platens. They are also used extensively to remove moisture from materials such as drying glue on envelopes as shown in *Fig.* 4, or to maintain the proper working temperature for materials being worked from hoppers. Steel sheath heaters are designed for sheath temperatures up to 750 F maximum while stainless steel sheaths are supplied for temperatures up to 1200 F. These heaters are 5/16-inch thick and 1½ inches wide with both terminals located at one end or with one at each end. They are standard in lengths of 8, 12, 18, 24, 30, 36 and 43 inches, in wattage ratings from 150 to 1500 watts at 115 or 230 volts. Special wattage ratings can also be made from 50 to 3700 watts depending on the length of the heater.

Strip heaters are designed for fastening to the bottom surface of a platen by clamps spaced 4 to 6 inches apart along the length of the heater. A simple clamp may be made from a 4-inch length of 1 by 2-inch channel iron with a hole in the center for a ⅜-inch diameter mounting bolt. Such a clamp will hold down two heaters at one point on their length, at the same time allowing them to be spaced one inch apart. If thermal insulation is to be applied, an air space of one inch on the outside surface of the heater sheath should be allowed.

Knowing the temperature of the material being heated, it is a simple matter to estimate the heater sheath temperature and thereby determine whether steel or stainless steel sheathed heaters should be used. Also, whether the rating selected will exceed the safe operating temperature of the heater. This is done as follows: First; obtain from the accompanying tabulation the watts per square inch of heated surface for the heater selected. Second; inasmuch as tests have proved that the temperature difference between the heater surface and the heater sheath is approximately 14 F per watt per square inch for a smooth and flat (machined) contact surface and 19 F for a rough and poor

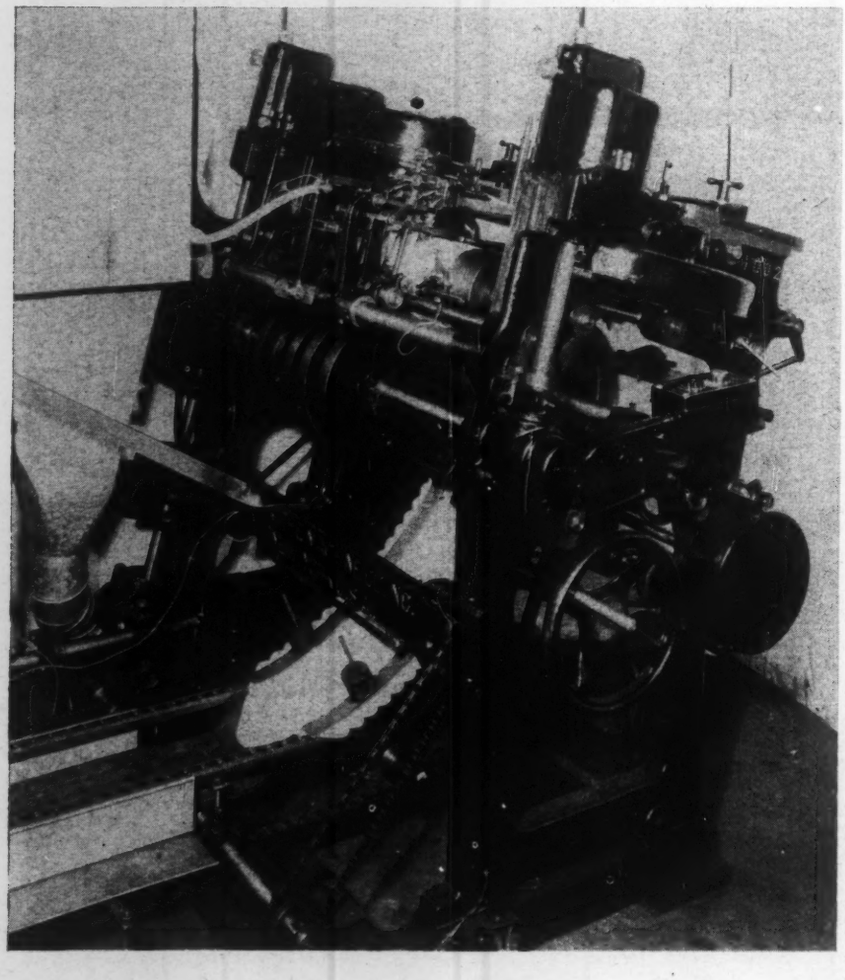


Fig. 4—Envelope machine utilizes strip heaters for air heating to dry glue on the envelopes as they pass through

contact surface it will be necessary to determine the type of surface involved. Third; with this information, the total temperature difference can be calculated by multiplying the watts per square inch of heater surface by the temperature difference per watt per square inch for the surface involved. Fourth; this temperature difference, plus the process operating temperature should be less than the sheath temperature. If this result is a temperature above 750 F a stainless steel sheath must be used.

For applications where it has been found that the sheath temperature of a strip heater would exceed 1200 F, a tubular element may be considered. This element is best suited for heating high-temperature platens because it lends itself to casting into the platen. Cast-in elements give excellent thermal contact between the heater sheath and material being heated with a resultant low temperature difference between the two parts. Tubular heaters are made in various diameters from 0.330 to 0.625-inch

Wattage Density or Watts per Square Inch of Contact Surface

Length of Heater (inch)	Steel Sheath* (watts per square inch)	Chrome-Steel Sheath† (watts per square inch)
8	25	42
12	21	29
18	16½	24
24	16½	25
30½	19	25
36	21	31
43	21½	25½

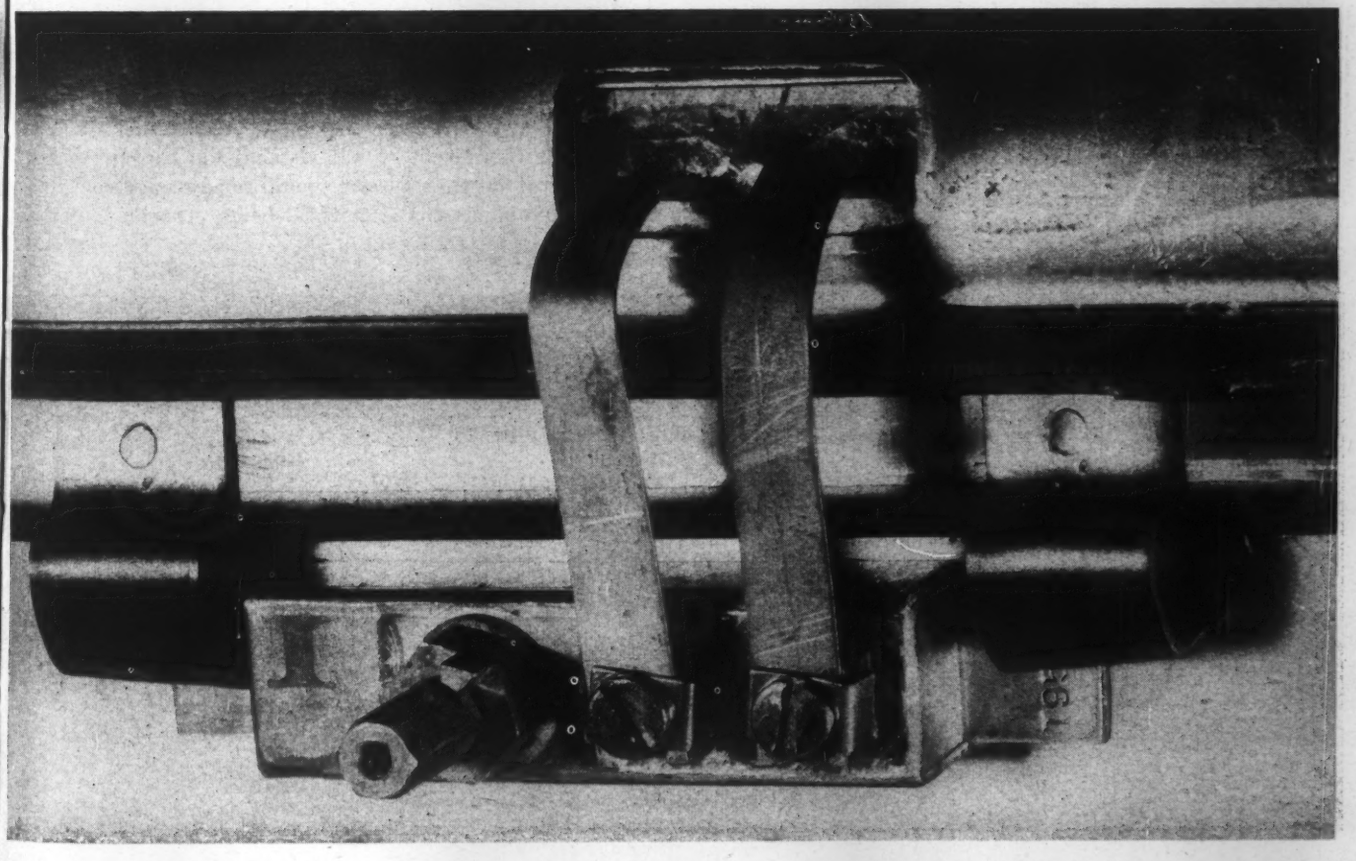
* Steel sheath maximum surface temperature 750 F.
† Chrome-steel sheath maximum surface temperature 1200 F.

Fig. 5—Narrow differential thermostat with a bimetallic strip for its thermal-responsive element applied to a mica-insulated heater. Thermal strip carries heater current

and in practically any length and rating desired.

Each manufacturing process for which controlled electrical heat is being considered has certain temperature limits within which the thermostat must maintain the element supplying heat to the process. For reasons of initial cost, overall simplicity and long thermostat and heater life, it is important that temperature limits no closer than the process actually requires be specified. Close thermal control requires a thermostat capable of operating with little difference between its opening and closing temperature. Such a thermostat is referred to as a narrow-differential thermostat and generally uses a bimetallic strip as its thermal-responsive element. Changes in calibration throughout the life of the thermostat are caused by contact wear which occurs due to contact arcing. The useful life of a thermostat is measured in terms of cycles of operation and the rapid cycling required for narrow differential control will, therefore, affect its useful life.

Illustrated in *Figs.* 5 and 6 is a type of thermostat designed especially for the narrowest temperature differential control that is possible with a simple bimetallic device. Sensitivity of this thermostat is due partly to the fact that heat is generated in the bimetal by the heating element current. This localized heat acts as a cut-off anticipator and causes the thermostat to open the circuit somewhat before it would if the opening depended only on the heat supplied to the bimetal by the heating element. Temperature undertravel is also reduced because the localized heating stops at the instant the thermostat opens and thus allows the bimetal to cool rapidly and reclose the circuit. In a thermostat of this type, the bimetal motion is not restrained by springs or other members and, therefore, lends itself to narrow differential temperature control. This thermostat has a maximum current rating of 10



amp at 115 volts or 8 amp at 230 volts, ac only. At maximum current the thermostat has an average useful life of about 250,000 operations and will operate about 30 times per hour in an average application. If the load is less than the rated current, the life will be increased greatly. For loads greater than the thermostat rating, a relay should be used or the thermostat connected in series with the coil on a magnetic contactor. This thermostat can be obtained with a factory calibration at any temperature up to 600 F or can be obtained in an adjustable type with standard ranges of 32 to 500 F or 250 to 450 F.

For reasons of strength and convenience, the mass or thermal capacity of die blocks and platens is frequently considerably greater than the thermal capacity of the material being processed. When this is true the no-load or idle periods, during which material is not being processed, require very nearly the same wattage input as do the periods during which material is being processed. Under these circumstances, it is not necessary to mount the thermostat at a point immediately adjacent to the area from which heat is being drained because the thermal capacity of the tool acts as a ballast to hold the working area at a constant temperature between no-load and full-load conditions.

High thermal capacity also necessitates a relatively large cross-sectional area of good heat-conducting material. It is then possible to mount the thermostat on any convenient outside surface and thermal insulation need not be applied to the thermostat unless abnormal air currents are passing over it.

In applications where the parts being processed have a relatively large thermal capacity such as when liquids are being heated or evaporated, it is necessary to locate the bimetallic element of the thermostat in very good thermal relation to the working area. A thermostat which lends itself to this type of mounting is shown in *Fig.* 6. Design of this thermostat is such that the bimetal does not carry the current, also that the bimetal is slightly loaded by having its motion resisted by a contact carrying spring member. These two factors cause this thermostat to produce a definite difference between opening and closing temperatures as measured at the controlled surface. This difference is commonly referred to as the thermostat differential. Under idle conditions, this differential on an average application is approximately 25 F but will be substantially reduced under full-load conditions because the thermostat will immediately respond to a temperature drop at the working surface.

When carrying full-load of 15 amp at 115 volts and 10 amp at 230 volts, ac, the useful life of this thermostat will be about 250,000 cycles of operation. For average conditions with a differential of 25 F it will cycle 10 to 12 times an hour and will, therefore, have an operating life of about three times that of the previously discussed thermostat. It may be calibrated to any temperature up to 650 F or may be adjustable, having any temperature range desired.

Should Be Close to Working Surface

This thermostat can be placed in good thermal contact with the working surface by mounting it at a central point on the tool surface opposite the working surface. Thickness of the metal making up the working surface should be minimum at the point where the thermostat is mounted. Sometimes this can be done by casting or machining a cavity in the back surface to receive the thermostat and the thin metal section thus created between the thermostat and the working surface will permit the rapid response of the bimetal to temperature changes at the working surface. With correctly applied heaters, the heat transfer from the element to the thermostat will also be good and will aid in reducing the temperature differential of the working surface.

A thermostat should never be located at a point which places the heating element between the thermostat and the surface to be controlled. Such an arrangement would cause the thermostat to be ineffective because the heating element would act as a thermal insulator between the thermostat and the working surface and cause a temperature lag between these two points. Also, under this condition, the thermostat would respond too rapidly to heat supplied by the element and would not be affected by the heat requirements of the material being processed.

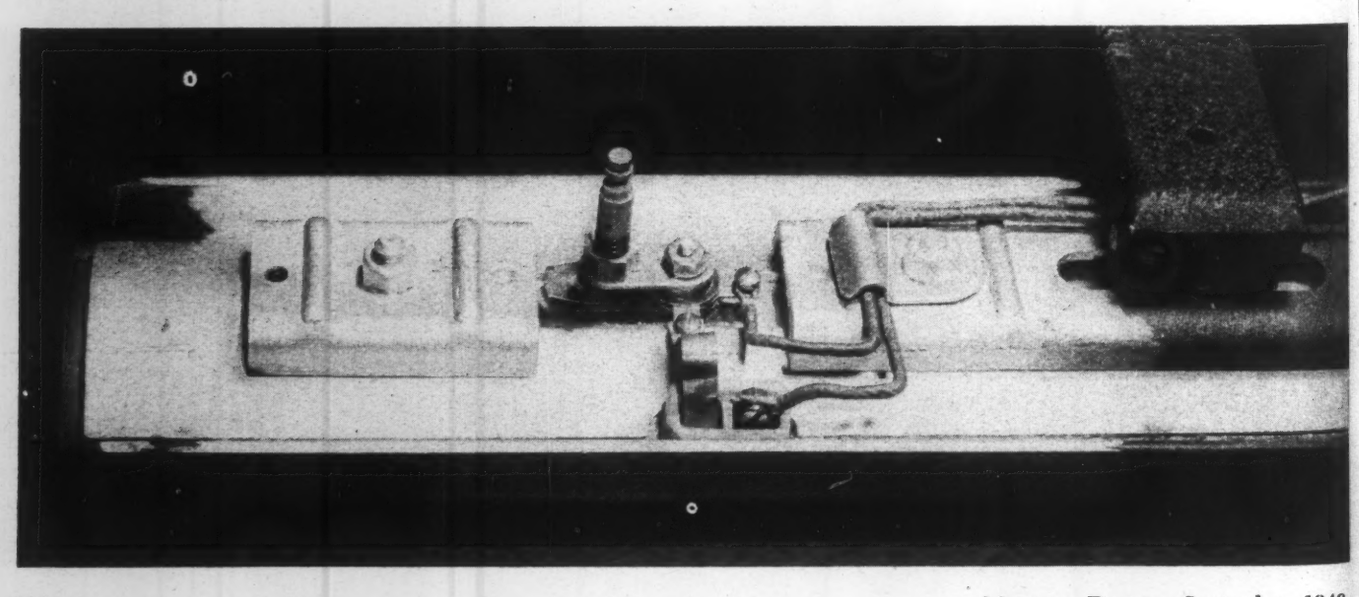


Fig. 6—Bimetallic element has been designed to maintain good thermal relation with working area when mounted as shown. Heating current is not carried by thermostat

Mechanical Seals

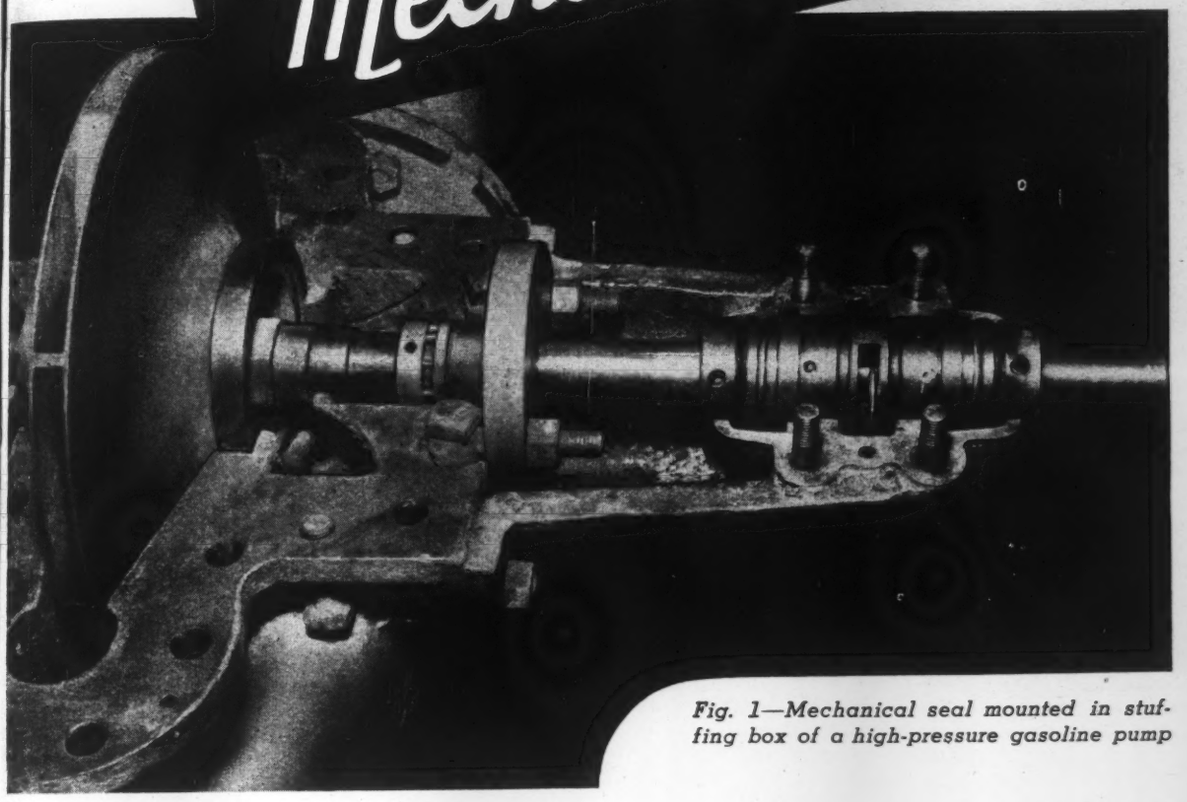


Fig. 1—Mechanical seal mounted in stuffing box of a high-pressure gasoline pump

Part I–Basic Considerations

By Douglas R. Lewis
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WHERE mechanical seals have been properly designed, selected, and installed, results have proved so effective that they preclude any return to packed boxes on rotating shafts. It is true that there have been many failures of seals but, in nearly every instance, trouble might have been avoided by more careful attention to the problem. Seals are far less rugged than packing. They involve fine surface finish, close tolerances, delicate membranes, brittle materials, special lubricants, and good bearing design. In some instances the seal manufacturer has been at fault in not confining his claims to what he knows to be true, or in not selling his device on a frank experimental basis. A typical mechanical seal for high-pressure work is illustrated in *Fig.* 1.

There is not enough experience data at this stage to warrant unreserved prediction of success on all applications, and it follows that unless one has an actual performance background he cannot be certain that the seal selected will do the job. Service in identical conditions gives such a background but sometimes a seemingly unimportant factor will mean

ROTARY SEALS must be selected and applied carefully if satisfactory performance is to be achieved. To assist the reader in appraising seals this series of articles will deal with the design problems involved, types of seals for specific applications and theoretical seal design

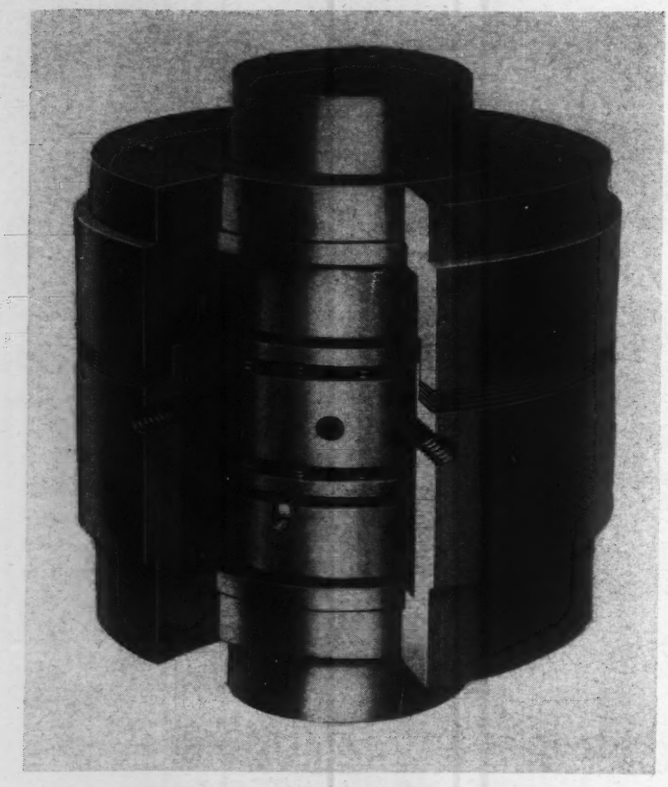


Fig. 2—Special seal for use on high-pressure pumps

the difference between success and failure. Tests should always be made in actual use and, if that is not possible, laboratory test conditions should duplicate those in the field as nearly as possible. The following case might serve to illustrate this point.

Mechanical seals were sold for a line of gasoline pumps to be used on shipboard, the seal being somewhat like that shown in *Fig.* 2. The seal manufacturer was told that a positive pressure would always be imposed on the seal. In actual test at the pump plant the pressure was positive and the seal was perfect. In use it overheated, used large quantities of grease, scored rapidly, and leaked whenever the pump stopped turning. A compound gage put on the suction side of the pump showed a pressure of about 14 psi when the pump was idle, but when the pump went to work the gauge dropped to 19 inches of vacuum. The many twists and turns of the suction line to the pump caused so much friction that the pump was effectively throttled. Thus, the seal was called upon to keep air out of the pump instead of to keep gasoline in it. The particular type of seal could have been designed for either function but not for both. Two holes drilled in the casing to allow discharge pressure to enter the box took care of the difficulty.

No Seals Have Universal Applicability

Some seal manufacturers, recognizing that failure to analyze each application thoroughly can lead to much grief, offer a complete engineering service. A prospective user should never hesitate to make use of such assistance, for he will be time and money ahead as a rule. A great deal of advantage lies in finding out by reference to the manufacturers' experience what can be done, but more advantageous is the knowledge of what cannot be done. As the art progresses it is becoming more and more clear that particular types of seals are superior in certain respects but that none has universal applicability. It may be that some problems are so special that no existing seal presents a solution, thus necessitating a change in design of the equipment to be sealed or even indicating the development of one's own seal. By and large, however, it now appears possible to seal mechanically rotating shafts in nearly all industrial fields and, if the application is correct, long and trouble-free service can be expected.

In discussing the problems of sealing, a brief analysis of the action of the stuffing box follows to serve as an introduction. There can be little doubt that progress in the art of sealing rotating shafts has suffered because the reciprocating rod stuffing box was carried over for rotating

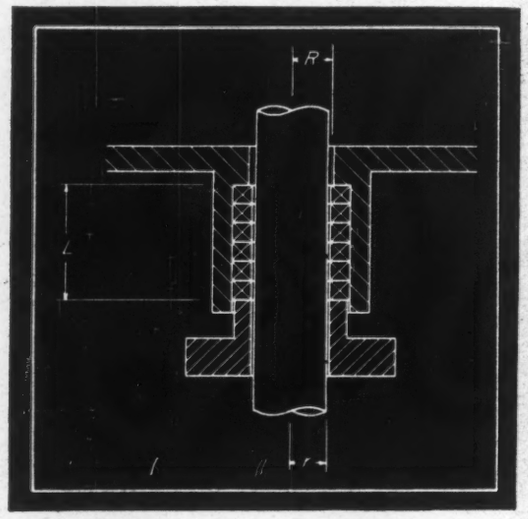


Fig. 3—Schematic for a stuffing box indicating measurements for calculating leakage

shafts. However, it is not difficult to understand the adaptation of the stuffing box to rotating shafts because the idea of packing is fundamentally sound. A cylinder made up of rings of a comparatively inelastic substance is interposed between two fluids and is then compressed around the connecting shaft until the space between its inner surface and the shaft is small enough to prevent the intermingling of undesirably large quantities of one fluid with the other. One of the fluids is generally the atmosphere surrounding the machine. The rate of flow through this cylindrical orifice, *Fig.* 3, can be calculated from:

$$Q=\frac{\pi P}{8\mu L}\left(R^4-r^4\right)-\frac{(R-r^2)^2}{\log_e(R/r)}$$

where Q = rate of leakage, P = pressure drop, L = length of packing along the shaft, R = inside radius of packing ring, r = shaft radius, and μ = fluid viscosity*.

Application of this equation is somewhat complicated by the fact that R is not easily determined, and also by the fact that in most stuffing boxes it varies because the packing rings at the box bottom are compressed less than those

* For a simplified version of this equation see "Leakage in Capillary Seals of Hydraulic Valves and Pumps"—Paul G. Exline, *Product Engineering*, April, 1946, Page 290.

next to the gland. The quantity r, also, will not be uniform on scored shafts.

It is interesting to note that speed does not enter into the calculation. But motion should not be disregarded, because it usually introduces eccentricity which in turn affects the net result. Most important is that Q increases with R. Therefore there will always be an amount Q except in the case where $R = r$. This condition is virtually impossible, but it is approached as $R - r$ approaches zero. In other words, to make Q very small the packing must be drawn up tightly in order to reduce R to the minimum. The well known results of such an operation preclude its use among careful maintenance men who wish to avoid scored shafts or sleeves (*Fig.* 4), dangerous temperatures, short packing life, overloaded drivers, and increased power costs. In short, there are few who will not agree that suitability of packing on a rotating shaft is directly proportional to the permissible leakage.

However correct in theory packing may be, therefore, practical considerations disqualify its use wherever permissible leakage is zero. The no-leakage requirement may arise from one or more circumstances, most of which arise in turn from comparatively modern developments. Until corrosion-resistant alloys became available, for example,

Fig. 4—Scored shaft sleeve resulting from packing

corrosive chemicals were not pumped, simply because there was nothing from which suitable pumps could be made. Chemical pumps for nearly all fluids are now standard equipment. That the seals on such pumps should be leak proof goes without saying. In the same class is equipment for working toxic or inflammable substances. Manufacturers of refrigeration machinery recognized early the inherent weakness of packing for such fluids as ammonia, sulphur dioxide, methyl chloride, and other refrigerants.

Another important factor, high cost of the fluid, also had to be given consideration. Freon, at something like eighty cents a pound, could not be wasted through a stuffing box. The writer recalls watching a test of vacuum pumps for a recent war project in which loss by drops of a special lubricant worth $2000 a gallon was the deciding factor; and there is a point in a recently developed vitamin process where a concentrate must be pumped which is worth more than its weight in gold.

Hazards Must Be Considered

Factors of corrosiveness, toxicity, inflammability, and high product value are all obvious. There are more subtle ones, which may not occur to the designer or user. A seal which allows a nauseous fluid to escape is undesirable for reasons of comfort and morale. If a fluid is slippery like oil and is not adequately piped away, it presents a hazard to safe walking or working around a machine. Materials which crystallize, such as sodium phosphate, or which harden to viscous films, such as butyl acetate, cannot be piped away and must be scraped or flushed away with a solvent periodically. Then too, there are fluids such as propane or butane under high pressure, which, upon escaping through the small orifice, absorb heat from surrounding substances as they expand, causing the condensed moisture in the air to freeze and perhaps lowering the viscosity of nearby lubricants so that bearings have short lives.

It is also possible, as a rule, that elimination of the stuffing box may allow better design by reducing the shaft overhang or diameter, permitting the use of smaller bearings, or obviating cooling jackets, sleeves, lantern glands, extra piping, internal passages, etc. On certain types of machinery, packing a stuffing box becomes a major operation because of heavy parts or inaccessibility.

Most important consideration in many instances where the decision to be made is whether or not there shall be a stuffing box is the matter of cost, not only in fabrication of the machine, but also in operation and maintenance for better sales appeal. In considering this factor a chart such as is shown in *Fig.* 5 may be helpful. The cost ordinates are based on figures for replacement of worn parts,

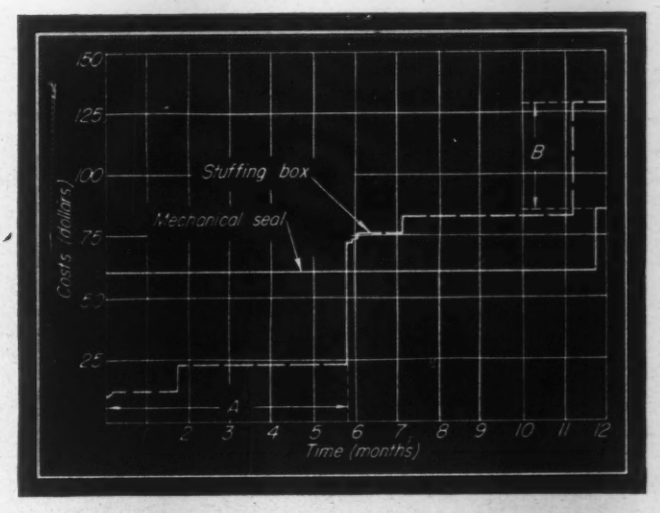


Fig. 5—Comparative costs of operation for a mechanical seal and stuffing box, average conditions. Time required for seal to pay for itself is shown at A, and saving at time seal is replaced is shown at B

labor, loss of product, auxiliary equipment, power, down time and, in rare instances, high insurance rates.

Packing for a pump, for example, may cost $10. After it has run a day a man at $1.50 an hour may take half an hour to adjust it. In about three days he may take it up again, and in a week once more. In a month he may find in necessary to add a ring of packing costing $1.50. In some plants, safety regulations require the insertion of blanks between shut-off valve flanges before installing a packing ring.

In another four months or so the sleeve must be renewed, with all the incidental labor charges plus say $15 for the cost of a new sleeve. If the sleeve and impeller are integral the materials cost might be $40. A new set of packing is usually installed with a new sleeve, which costs $10 more, and the whole process begins again. During all this time there are such constant costs as loss of product and the cost of the power required to wear out the parts.

The unpacked box, on the other hand, has a comparatively high initial cost, say $60, but once put into operation its maintenance cost is low. The point of intercept between the two cost lines, in *Fig.* 5, shows how long it takes the unpacked box to pay for itself, and the vertical distance between the two lines at the point where the unpacked box requires replacement of worn parts shows how much has been saved. Whether the time required for the unpacked box to pay for itself is short enough to justify the initial cost is a matter for special judgment in each case, but it is almost always possible to show a saving eventually. In fairness to those who hold a brief for packing it should be pointed out that the case against the stuffing box is not always as bad as outlined. There are many times, though, when it is much worse.

Mechanical Seals Have Basic Advantage

As wear takes place in a packed box, either on the packing or on the sleeve or shaft, the interstice increases in size until the rubbing pressure is not great enough to cause further wear. No harm is done, of course, if the point at which wear ceases allows no more than permissible leakage. For reasons pointed out earlier, however, it may be necessary to prevent any leakage at all, or at least to prevent as much leakage as would result from the increased size of the orifice due to initial wear. It then becomes necessary to bring the interstice back to minimum size by tightening the gland. But the interstrice is soon enlarged again, requiring repeated tightenings. *Fig.* 6 shows a sleeve worn through by periodic tightening of the gland.

Sealing Surfaces Should Be Normal to Shaft

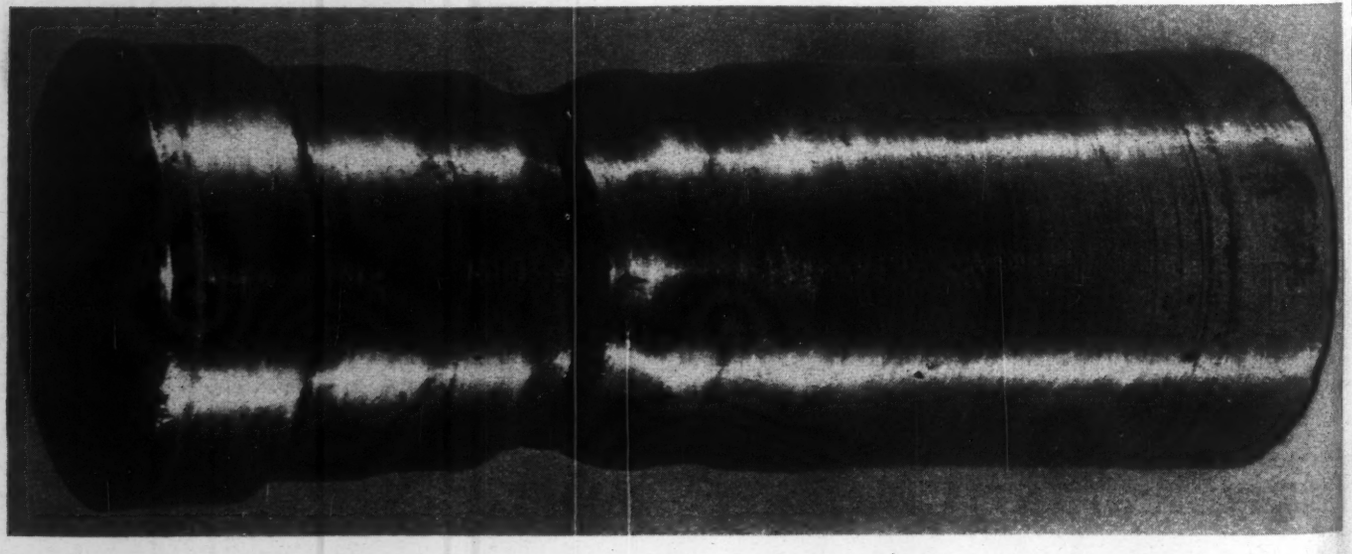
It is apparent that if the plane of friction could be placed at right angles to the shaft instead of in a cylindrical plane parallel to the shaft contour, wear might be compensated for by a unidirectional force parallel to the axis of rotation. This, in essence, is what a mechanical seal does, although the friction plane may depart slightly from normality to the shaft and may even be curved. The force may be applied by a spring, a resilient rubber-like material, or by effective areas against which the fluid pressure works.

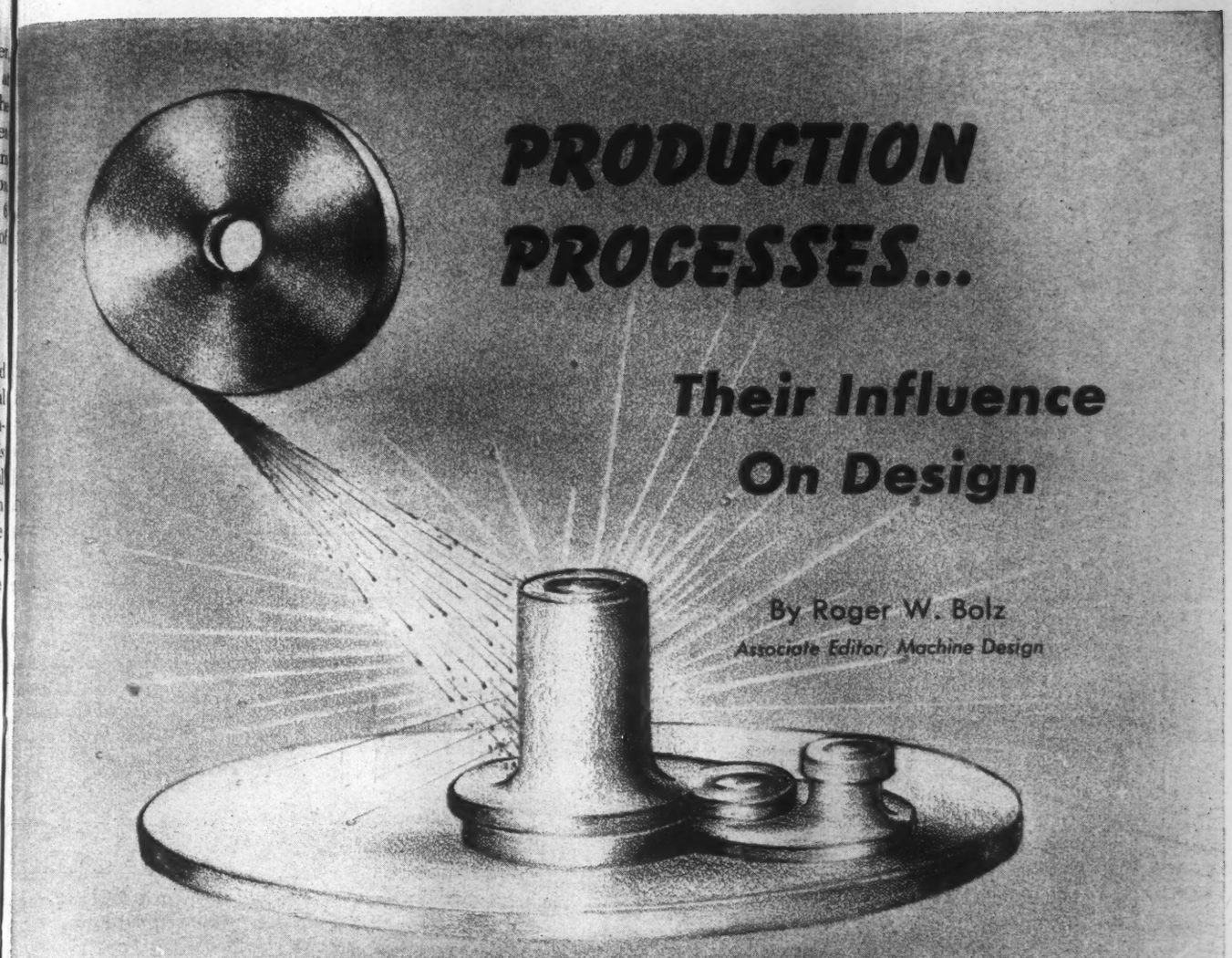
Rubbing faces may be disposed about the shaft so that tendency to leak is radially inward across them or radially outward. The axially movable member may be the rotating or the stationary member or both. The contact pressure may be balanced so that it is not a function of the fluid pressure, or it may have varying degrees of unbalance. There are various methods of attaching the parts to the shaft or housing. As new materials with favorable mechanical and chemical properties are developed new seals may take advantage of them.

Many combinations are possible and inventors are steadily producing them. During the last year an average of two patents a month have been granted on mechanical seals. There are at present at least twelve companies who make seals as a standard line and many more who make them for use in their own equipment exclusively. One or the latest designs, *Fig.* 2, is a stuffing box and seal combined, made as a complete unit by the seal manufacturer and ready to be bolted to the pump which is made without a box.

Succeeding articles in this series will discuss types, installation precautions and theoretical seal design.

Fig. 6—Sleeve worn through by packing gland





PRODUCTION PROCESSES...

Their Influence On Design

By Roger W. Bolz
Associate Editor, Machine Design

Part XV—Shot Peening

USE of surface compression to improve strength is undoubtedly ages old and was practiced as cold hammering by old Toledo swordsmiths, for example, to achieve phenomenal results. Benefits derived from such cold working of metals, often incidental with many manufacturing processes such as rolling, heading, bending, stretching, etc., consist mainly of increased fatigue strength imparted to machine parts when their metal structure is stretched, twisted or compressed beyond the elastic limit.

Selective surface compression imparted by a rain of metal shot impelled either from the rotating blades of a wheel or by air blast achieves like beneficial results on a great variety of machine parts such as crankshafts, springs, shafts, gears, torsion bars, connecting rods, etc., *Fig.* 1. Each metallic shot has the effect of a tiny peen hammer to distort the surface structure of the metal and set up compressive stresses. This compressively stressed layer usually extends 0.005 to 0.010-inch below the surface and is balanced by residual tensile stresses within the interior.

Inasmuch as most fatigue cracks have a tendency to start at the tension point of a stressed section, a compressively stressed outer layer at such a point will act to reduce the net applied working stress. Cold work thus increases the strength of the part but sacrifices a portion of the ductility. Intensity of peening therefore must be limited because an excess of working acts to reduce the resistance of a part under repeated stress by exhausting the ductility and producing incipient cracking. As little as 3 or 4 per cent ductility in a machine part after assembly, however, is considered sufficient for all probable readjustments of stress distribution.

Although the equipment utilized for shot peening is essentially the same as that used for blast cleaning, it necessarily requires more accurate control and auxiliary equipment for consistent results. In rare cases blast cleaning of certain parts results in an increase in fatigue strength but more often the opposite is the case, *Fig.* 2. Increased fatigue life has been achieved with certain aluminum wing spars sand-

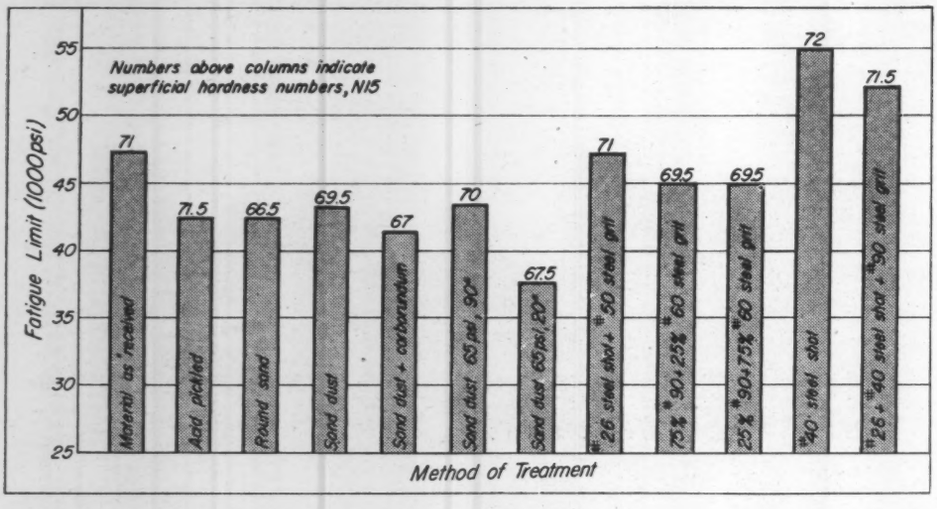


Fig. 1—Above—Arrangement for production shot peening Allison engine connecting rods at Cadillac Motor Car Co. plant

Fig. 2—Left—Graph of fatigue limits resulting from various methods of surface treatment using a specimen of 0.054 per cent carbon steel heat treated to 115,000 psi tensile. From "Prevention of Fatigue of Metals"—Battelle Memorial Institute, John Wiley & Sons, Inc., New York, 1941, Page 59

blasted for finish, but most all investigations of ordinary sand blasting show no appreciable increase.

The important difference therefore lies in the type of blasting material and its condition. For successful peening, whole unbroken shot is necesssary. Broken shot, having a much lighter mass, gives a lower or varying peening intensity and inconsistent results. In addition, the sharp edges of broken shot pit and abrade the surface creating scars which act as stress raisers with a resultant decrease in fatigue life. Production peening equipment thus requires continuous removal of broken shot and a rationing system to introduce new shot at the same rate at which the fines are removed.

Since it is usually desirable to peen specific portions of a part, specially designed machines are most practical in production peening. In addition, accurate control being a prime necessity for optimum results, closely regulated intensity and coverage are required. The wheel impeller machine has been found best suited to give these results at minimum cost since velocity and distribution can be closely governed. A few of the great variety of special handling methods are shown in *Fig.* 3. Air operated blast machines are much more costly to operate but are sometimes advantageous in that the target space can be varied by changing orifices and the shot stream can be moved if desired. On occasion a combination of the two methods can be used but in general the compressed air method would only be desirable where the work to be peened presents only a small area or one not possible to reach with a wheel stream.

Intensity of peening is an instantaneous condition of the blast itself and depends upon the velocity, hardness, size and weight of shot, and duration of peening. To date no quantitative rules exist for assigning values to these factors so as to produce the desired results. However, the Almen gage—a strip "A" for low intensities, ¾-inch wide by 3 inches long and 0.051-inch thick (±0.001-inch), and one "C" for high intensities, ¾-inch wide by 3 inches long and 0.0938-inch thick (±0.001-inch), both with a rockwell C hardness of 44 to 50—is used to measure intensity and obtain uniformity. These are placed on a dummy part, *Fig.* 4, at the critical positions and subjected to the same conditions of peening as that to be used for production. The curvature produced on the sample—convex surface

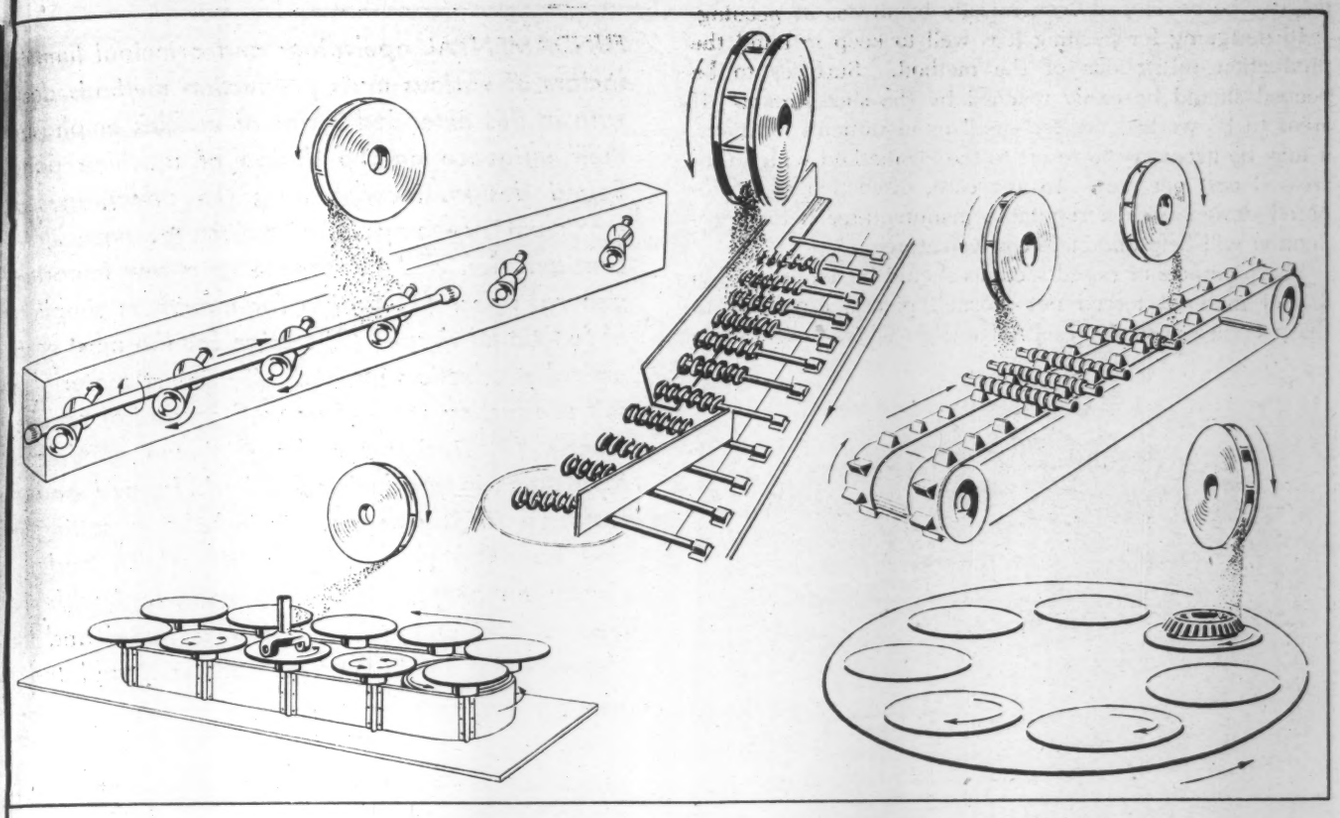
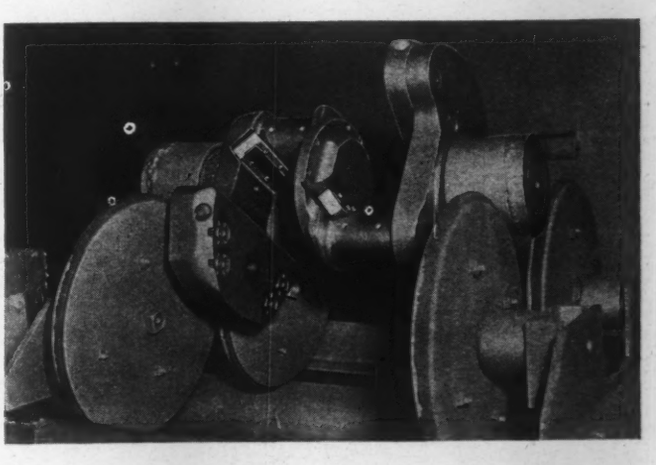


Fig. 3—Above—Schematics of a few of the many production arrangements for peening machine parts

Fig. 4—Below—Dummy crankshaft fitted with Almen test strips to determine peening intensities in critical areas



in the peened side—is measured as an indication of the intensity. Proper intensities are for the most part a result of experience, judgment and actual laboratory fatigue tests.

Shot peening can be applied to a great many irregular shapes which cannot be rolled or otherwise treated. Peening can be applied to finished parts or to specific areas of highly stressed parts such as shaft fillets to offset stress concentrations. Peening in general tends to offset or delay pitting corrosion with its resultant fatigue cracking. Peening applied to gear teeth causes no appreciable distortion and produces tooth surfaces highly resistant to pitting corrosion under localized heavy stresses at bearing points with improved wear resistance, *Fig.* 5. Records show that gear life improvements as high as 900 per cent have been achieved.

Used in place of polishing, a peened surface finish often reduces production costs considerably while at the same time imparting a useful gain in fatigue strength. It can also be utilized to increase resistance to surface damage such as light bruises, "fretting corrosion", pitting corrosion and surface decarburization.

In general, although peening will increase the fatigue life of any part subject to bending or twisting, it has little or no effect on axial push-pull or direct tension-compression stresses. Consequently, parts such as bolts or tie rods under axial tension only will not be benefitted unless they encounter twisting or bending stresses due to load or vibration. Torsion bars on the other hand will be tremendously benefitted. If unskillfully applied, peening may actually weaken a part rather than strengthen it. Also a further limitation is presented in that the thickness of the peened skin, being only on the order of 0.010-inch in thickness for normal pieces, cannot be expected to achieve the same improvements in large, thick parts as has been the case with so many thin pieces. Finally, shot peening cannot obviate heat treatments necessary to secure desired properties but is a useful adjunct to produce increased fatigue life.

DESIGN: In addition to combatting premature failure of parts such as gears and springs already beyond the design stage, peening permits the use of higher stresses on standard parts such as those in engines to achieve increased power without increased weight. In many cases the increase in fatigue life permits a reduction in size and cost of parts, especially where these are designed for a finite service life. By shot blasting, springs may be designed to the elastic limit since the problem will not be with breakage but loss of load due to the spring not returning to original length. Consideration should also be given to employment of lower cost steels in which the desired fatigue proper-

ties can be developed economically by means of peening.

In designing for peening it is well to keep in mind the production restrictions of the method. Surfaces to be peened should be easily reached by the shot stream. If areas to be worked are too small or in difficult locations, it may be necessary to resort to the air method with an increased cost per part. In any case, discussion of a proposed design with a reputable manufacturer before production will help eliminate costly features.

Sharp corners or raised sections should be avoided. Thin fins of material forced out at such places, especially at sharp corners, usually result in notches which act as stress raisers to offset most of the other advantages. These sharp fins can be avoided by chamfering the edges of the part before peening or by designing the peened portion with rounded edges, *Fig.* 6.

***FUNDAMENTAL** operations and principal limiting factors of various mass production methods dealt with in this extended series of articles emphasize their influence on the design of machine parts. Sound design, incorporating the advantages of maximum economy in manufacture, necessitates consideration and adherence to a few important general rules: (1) Design for maximum simplicity of functional shape; (2) Design for the most economical production method; (3) Select material to suit production method as well as design requirements; (4) Design for the minimum number of operations in producing finished, formed or cast surfaces; (5) Design for ease in holding, setting up and handling parts; (6) Specify finish and accuracy no greater than that consistent with the type of part or machine being designed and the production method or methods contemplated*

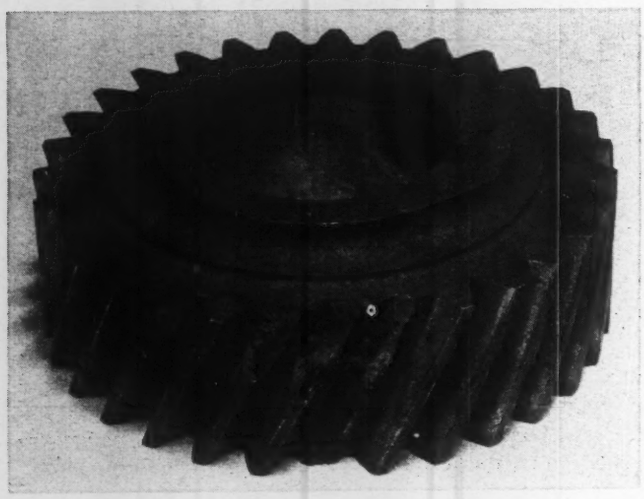
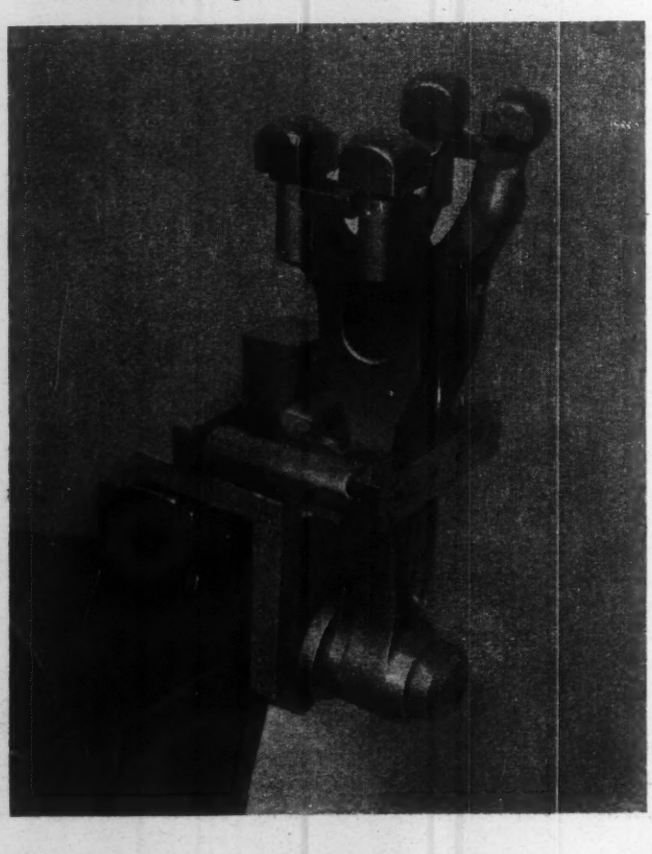


Fig. 5—Above—Typical transmission gear failure eliminated by shot peening with no change in original design

Fig. 6—Below—Peened portions require smoothly rounded or chamfered edges to eliminate sharp projections



As a rule grinding has been employed to avoid surface defects which normally contribute to low fatigue life of parts. With shot peening, however, grinding can often be eliminated with a resultant economy in costs. Where a peened surface is to be a bearing area, the peaks produced by peening are usually cut down by honing. The plateaus thus created with depressions between effect a better lubricated bearing surface wherein the pockets act as oil reservoirs.

MATERIALS: Shot peening can be used successfully on all steels including high-tensile steel alloys, cast irons, malleable irons, and on practically all nonferrous metals. Results obtained vary in degree with the cold working characteristics of the material and design of the parts. Naturally, those materials that are ultrasoft and consequently unaffected by cold working and those that are ultrahard provide little for improvement by means of peening. Material should be free from seams, cracks or inclusions or peening may be ineffective.

TOLERANCES: Practically all of the methods used to increase fatigue life affect in some way the size or shape of a part. Peening on the other hand has no appreciable effect on size and consequently can be utilized on finished parts without materially altering the original finished dimensions. As previously noted, gear teeth are peened without affecting operating characteristics other than to improve strength and wear resistance. Where honing is resorted to after peening, however, a small allowance should be made.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:
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(Figs. 1, 3, 4, 5, and 6) ..... Mishawaka, Ind.
Pangborn Corp. ..... Hagerstown, Md.
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Lubricants

...their proper specification and utilization in design

By Ben F. Hunter
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ECONOMICAL performance and life of a machine, regardless of its design, the materials employed in its construction, or the precision workmanship built into it, depends on a film of oil. Common practice in far too many instances is to design the more complicated machines to use several different oils and greases for lubrication. It is also common practice, and highly essential, for manufacturers of a given unit to recommend various types of lubricants as a guide for the ultimate operator.

Two general methods are used: Recommending oils and greases by brand name or by specifications based on ASTM standards. Such recommendations are usually a result of a study or tests or the known field performance of specific lubricants which the equipment manufacturer has found will provide the best lubrication for his machine.

In certain instances, the equipment manufacturer may

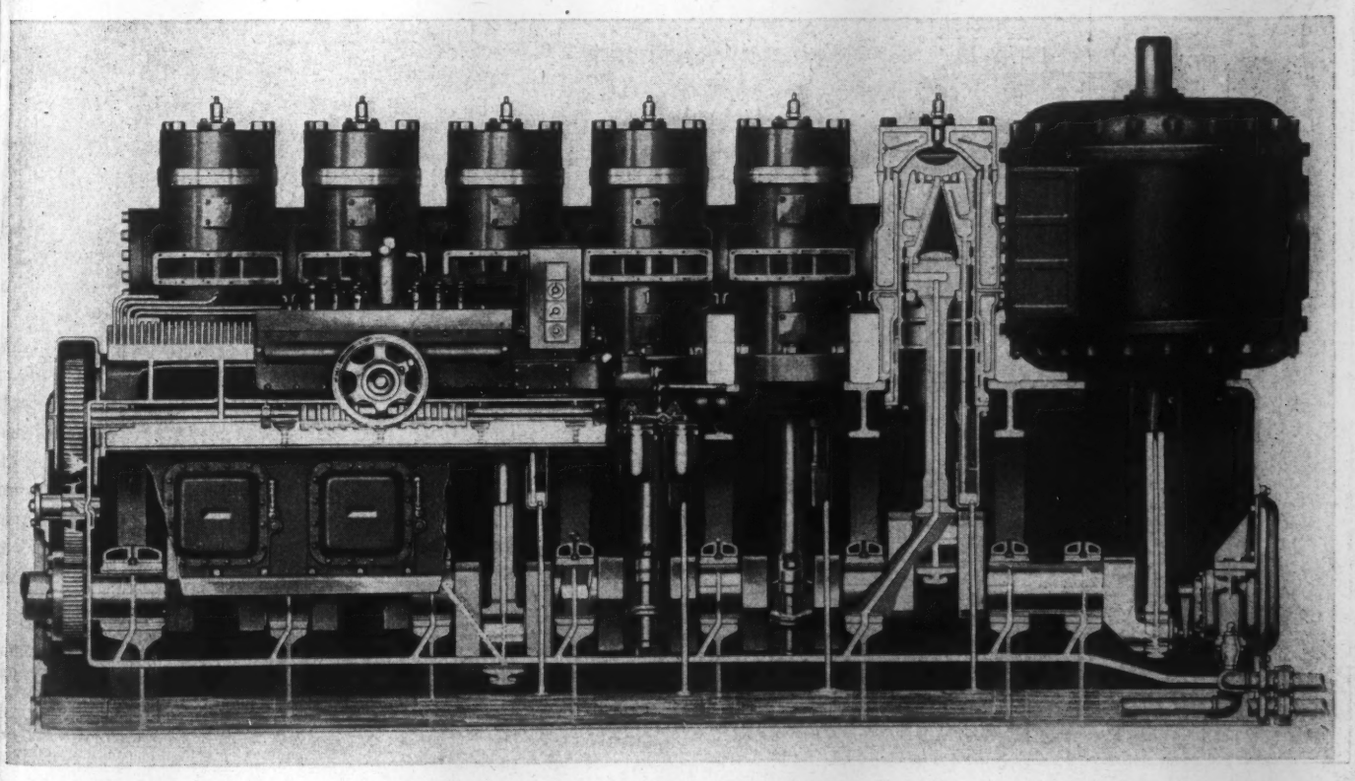


Fig. 1—Cutaway view of a Fairbanks-Morse diesel showing its lubricating system which provides a maximum of lubrication to all moving parts, wi tha minimum of attention

go so far as to insist upon the use of a specific lubricant if his performance guarantee is to be honored. Such practices are understandable but can present complications and difficulties for plant operators, since the particular lubricant recommended may not always be readily available. His regular supplier, however, may have an equally satisfactory product. The complications confronting the operator resulting from the recommendation of lubricants by a brand name become apparent in a major industry operating perhaps hundreds of different machines, each calling for specific brands of lubricants.

Should Use Minimum Number of Lubricants

In a recent survey of a large plant operating several hundred machines and machine tools it was revealed that 248 different oils and greases by brand name were recommended by the manufacturers of the different machines, where 22 different oils and greases would provide the maximum in lubrication. The greater the number of brands of lubricants in a given plant, the greater will be the possibility of applying the wrong lubricant to a specific machine or bearing and the greater will be the inventory cost for maintaining stocks.

Simplicity should be the yardstick in designing a lubricating system, utilizing the minimum number of different lubricants. The ideal machine uses one oil applied by means of a circulating system, with adequate oil purification equipment as a part of the unit, *Figs.* 1 and 2.

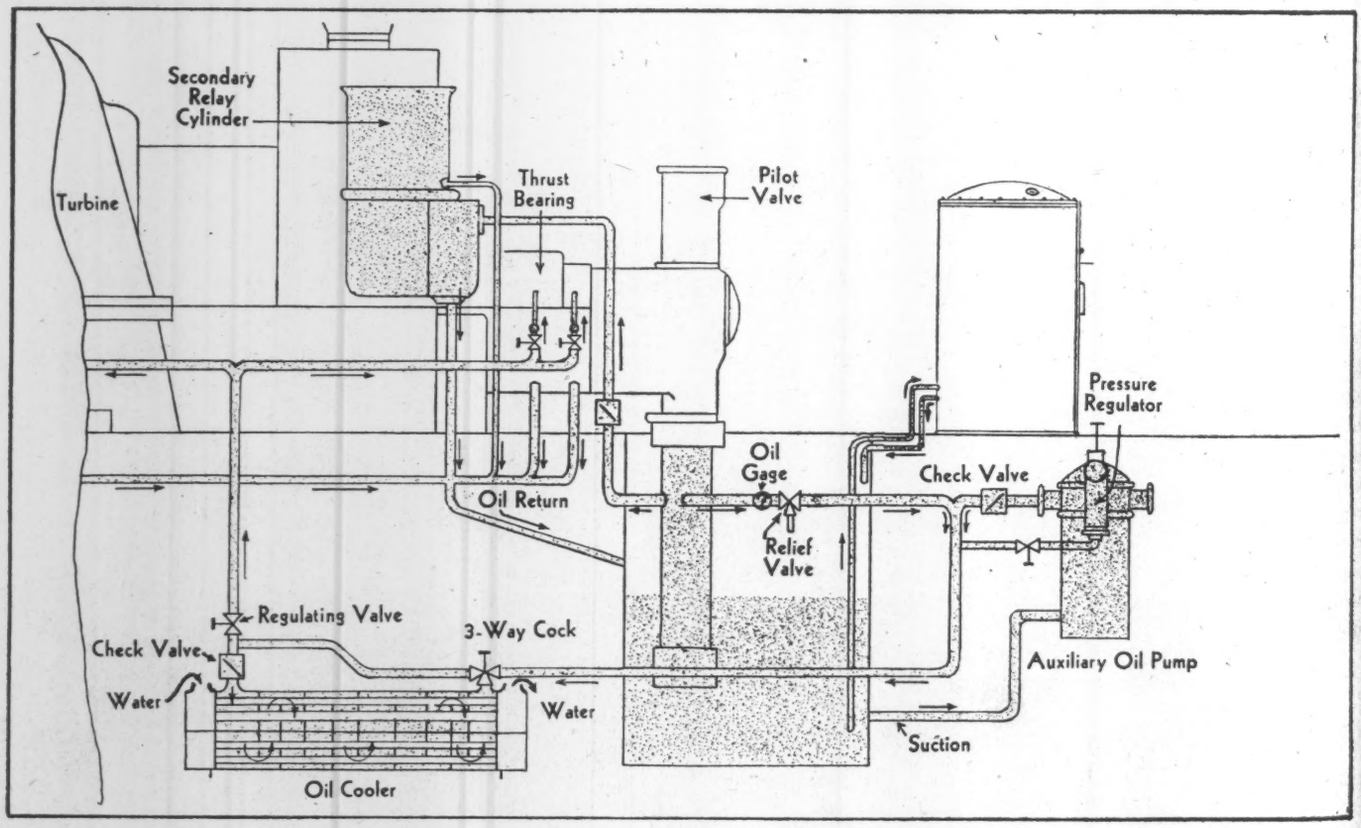
In many instances one oil will provide proper lubrication for a complicated machine if designers give due consideration to loads, speeds and temperatures of the various bearings or moving parts. For machines designed for grease lubrication, a centralized automatic system utilizing one grease would be the ideal. Where more than one lubricant is required for a given machine, separate circulating or bath lubrication should be provided for each lubricant.

Most satisfactory method of specifying lubricants for various machines is by specifications based on ASTM standards. The "viscosity" and "quality" of an oil and the "consistency" and "type" of grease are most essential. An outstanding example of the soundness of such practice is the automobile engine where viscosity in accordance with SAE numbers is the only recommendation made. Individual operators may then select the quality and brand in accordance with their own convictions. Another outstanding example is the steam turbine. Recommendations of viscosity, quality, and type of oil are the essential factors. Any reputable refiner or marketer of lubricants can supply an oil or grease that will provide proper and sufficient lubrication for any well-designed machine.

Hand application of lubricants to modern machines is obsolete and can only present operating difficulties and hazards which will eventually result in excessive wear on the moving parts, ultimate excessive maintenance cost, and shorter machine life. Modern machines should be designed around automatic constant lubrication, eliminating as far as possible the hazard of the human element.

Careful study of many expensive but essential machines now in service will disclose how most lubricating problems could have been eliminated on the drawing board. Machine designers with a basic knowledge of the fundamental theory of lubrication can go a long way in overcoming common lubricating problems, increasing production and improving plant operation with little, if any, added cost to the equipment manufacturer.

Fig. 2—Line drawing of an ideal circulating system applied to a steam turbine. Automatic lubrication provides for a copious quantity of oil to the bearings



Contrary to the belief of many, lubricating oils do not wear out but are rendered unsuitable for continued use by contaminants of foreign materials or organic compounds resulting from the oxidation of the oil itself. Every possible precaution should be taken in design to minimize contamination from foreign materials. The importance of temperature control becomes apparent when it is realized that the rate of oxidation doubles with each eighteen degrees (F) temperature rise. In other words, the useful life of an oil is cut in half or decreased fifty per cent with each eighteen degrees.

Products of oxidation of the oil itself act as catalysts to further increase the rate of oxidation; if these are allowed to accumulate in an oil to any appreciable extent (evidenced by neutralization number, say 0.50), the rate of oxidation may be many times that of the oil when neutralization number was, say, 0.10. Adequate purification to remove oxidation products as they are created may extend the useful life of an oil for an indefinite period.

Where two or more lubricants are required for a given machine, for example, a machine tool requiring a lubricating oil, hydraulic oil, and coolant, every precaution should be made in the design to prevent contamination of one oil by the other. While considerable progress has been made in this direction during recent years, there is still room for improvement on the drawing board.

Metals with which petroleum oils come in contact act as catalysts to accelerate the oxidation or decomposition of a lubricant in varying degrees. Listed below in accordance with the severity of their catalytic reaction with lubricating oils are metals commonly used in the construction of machines and coming in contact with the oil:

1. aluminum (least severe)
2. tin
3. iron
4. lead
5. zinc
6. copper (most severe)

Catalytic reaction increases proportionately with temperature rise. In design it is good practice to eliminate copper wherever possible from coming in contact with the oil, such as oil lines, oil tanks, heat exchangers, etc. This is especially important where oil temperatures are in excess of 150 F.

Additives Prove Helpful

During recent years, special lubricants, either oils or greases, or special compounds, and lubricating oils and greases containing various kinds of chemicals or chemical compounds generally known as "additives" have made their appearance on the market. Special lubricants or lubricants containing additives have unquestionably been helpful in many instances and their continued use may be expected. Unfortunately, however, no doubt due to the extensive publicity given them, they have been too generally regarded as a "cure-all" for all lubricating problems. Additives, depending upon the purpose for which they are intended, fall within certain classifications, such as:

1. **Oxidation inhibitors or oxidation stabilizers**—to retard the rate of lubricant oxidation
2. **Detergents**—to retain solids and products of oxidation and other contaminants in suspension
3. **Rust inhibitors**—to minimize corrosive effects of entrained water and, more recently, to prevent entrained water from corroding the oiling system or the parts of the machine with which it comes in contact
4. **Extreme-pressure lubricants**—to prevent galling and welding of bearing surfaces
5. **Antifoam agents**—to minimize foaming with resulting loss of lubricant
6. **Emulsifying agents**—to make oils soluble for coolants, etc.
7. **Wetting agents**—to improve oiliness properties
8. **Pour depressants**—to lower the congealing point
9. **Viscosity index improvers**—to provide a more constant viscosity with temperature change
10. **Metallic soaps**—to control solidifying properties of greases.

While special lubricants or lubricants containing additives have been a valuable contribution on the part of the petroleum refiners, technologists and scientists to industry, they are not recommended and cannot be expected to overcome lubricating difficulties attributed to improper design, incorrect methods of application of the lubricant, or contamination of the lubricant by foreign materials.

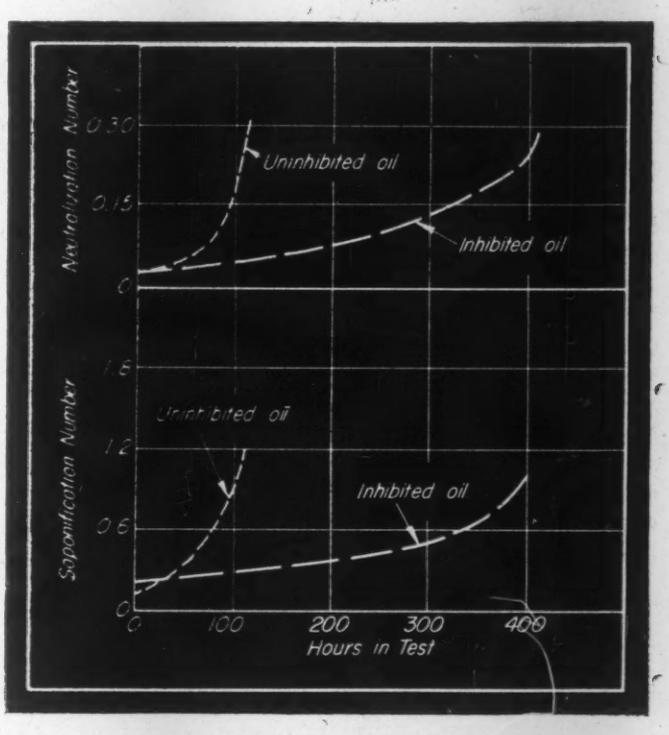


Fig. 3—Modified Staeger test on an inhibited and uninhibited oil. Neutralization number shows free acidity while saponification number shows total oxidation products that will react with potassium hydroxide. These curves are based upon data from Standard Oil Co. of New Jersey

Among the most popular and perhaps the most valuable additives appearing on the market during recent years are the first four in the tabulation, these will be discussed more fully in the following:

Oxidation Inhibitors: These are perhaps the most valuable contribution of petroleum refiners during the past decade. Better quality oils with reference to their rate of decomposition together with the oxidation inhibitors have provided lubricating oils whose useful life may be equal to that of a properly designed machine. This machine, however, must incorporate in its circulating system an adequate oil purifier to remove contaminants, either foreign or products of oxidation of the oil itself, as

they are created or find their way into the oiling system. All petroleum oils are oxidizable, to a greater or less degree, in accordance with their quality and the severity of service to which they are subjected. Under a given condition of service an oxidation inhibitor can be expected to prolong the useful life of an oil, but it does not follow that the life of the oil is indefinite when subjected to specific temperatures and undue aeration or excessive contamination. In general, there is a growing tendency on the part of industry to expect too much from improved lubricants. Most of the better grades of lubricants on the market today contain oxidation inhibitors, particularly those oils for use in circulating systems or bath systems where they are used over and over again. Curves in *Fig.* 3 show the relative effect of inhibitors with respect to neutralization and saponification numbers.

DETERGENTS: Lubricating oils containing detergent type of additive were primarily and originally perfected and recommended to improve the lubrication of small, high-speed diesel engines where excessive contamination of the lubricating oil resulted from the incomplete combustion of the fuel. Detergents, in general, are some form of metallic soaps with or without the addition of oxidation inhibitors, antifoam agents, pour depressants, or viscosity index improvers.

The function of the detergent additive is to maintain dust, dirt, free carbon, and products of oxidation in suspension in the oil, preventing their concentrating and settling out behind piston rings, cylinder heads, valves, screens, etc., and impairing the efficient operation of the engine. Detergent type oils have proved their value in the small, high-speed engine where the volume of oil in the crankcase is small and the drain periods recommended by the manufacturer are followed in accordance with the severity of the operation, minimizing the accumulation of excess contaminants and abrasives which can create excess wear on the moving parts if allowed to accumulate beyond certain limits.

Detergents Are Not a Cure-all

Like all valuable contributions, it is felt that detergents have been too generally regarded as a "cure-all" for all internal-combustion engine lubricating difficulties and that their use is being extended to applications beyond those for which the detergents were originally intended. In the large size, properly designed, low-speed diesel engine lubricating difficulties resulting from contamination of the lubricant by unburned fuel is less pronounced except where the engine is operated under severe, intermittent, or overload conditions. The volume of oil in the lubricating system of a large engine is such that frequent crankcase changes are impractical from the standpoint of economy. Contamination products of the diesel lubricating oils are foreign materials, abrasives, water, products of oxidation of the lubricating oil itself, dilution, and products of oxidation of the unburned fuel. Any or all of these impair efficient operation in direct ratio to the percentage and nature of contaminants present.

Basically, therefore, detergent oils are nonpurifiable oils but can be strained or filtered through fibrous type filtering media which will remove varying percentages of sludge, lacquers, tars, free carbon, and larger size abrasives, but will not remove oil soluble products of oxidation and the more finely divided solids or abrasives which can ultimately result in excessive wear of the moving parts, stuck rings, etc., if allowed to accumulate beyond certain limits.

Use of detergent oils in the large, slow-speed diesel engines is somewhat new and time alone will prove their suitability. Some operators of these engines report satisfactory performance using detergent oils; others have discontinued their use, reporting excessive wear on the moving parts and generally unsatisfactory performance.

RUST INHIBITORS: The so-called "rust inhibitor", to minimize corrosion from entrained water in circulating oil systems, is among the newer types of additives. Primarily, oils containing rust inhibitors were perfected to minimize the corrosive effects of entrained water in the oiling system of new turbine installations. Perhaps the most serious corrosion difficulty experienced in the steam turbine is that of the oiling system itself. The logical approach to this problem would be to fabricate the oiling system from nonferrous metals having the least catalytic effect on the lubricating oil rather than to place the added burden on the oil whose function is to lubricate the moving parts of the unit.

Plating Type Inhibitor Is Best

In general, it is agreed that the most effective rust inhibitor is the "plating" type. Such an additive is added to and carried in suspension in the lubricating oil which, when circulated for several hours, days, or perhaps a few weeks, will plate out on the surfaces of the metal in the oiling system or turbine proper with which it comes in contact, thus preventing water from coming in contact with and corroding the metal. Unfortunately, some of the most objectionable points of corrosion in an oiling system or machine are the points above the oil level where the oil does not come in contact.

Regardless of the effectiveness of the rust inhibitor, it cannot be expected to prevent corrosion of metals with which it does not come in contact. In general, the most effective rust inhibitors are of such nature or composition that they affect other desirable properties in an oil, such as steam emulsion number or water-separating properties, especially steam turbine or circulating system oil.

EXTREME-PRESSURE LUBRICANTS: The term "extreme-pressure lubricant" is generally applied to lubricants containing some form of sulphur compounds, which act as an antigalling or antiwelding agent. This term may be somewhat misleading in that it may indicate the load-carrying properties of the lubricant rather than its antigalling or antiwelding properties.

Extreme-pressure or antigalling lubricants were originally perfected for the hypoid gear but are now finding their way into many industrial applications. The original extreme-pressure lubricants were more or less corrosive to the highly polished surfaces of certain metals. This objectionable characteristic has long since been overcome, and it is doubtful if there is a corrosive type being marketed today. The corrosive characteristics of the original extreme-pressure lubricants became generally known and there exists today among many equipment manufacturers, as well as industry in general, the belief

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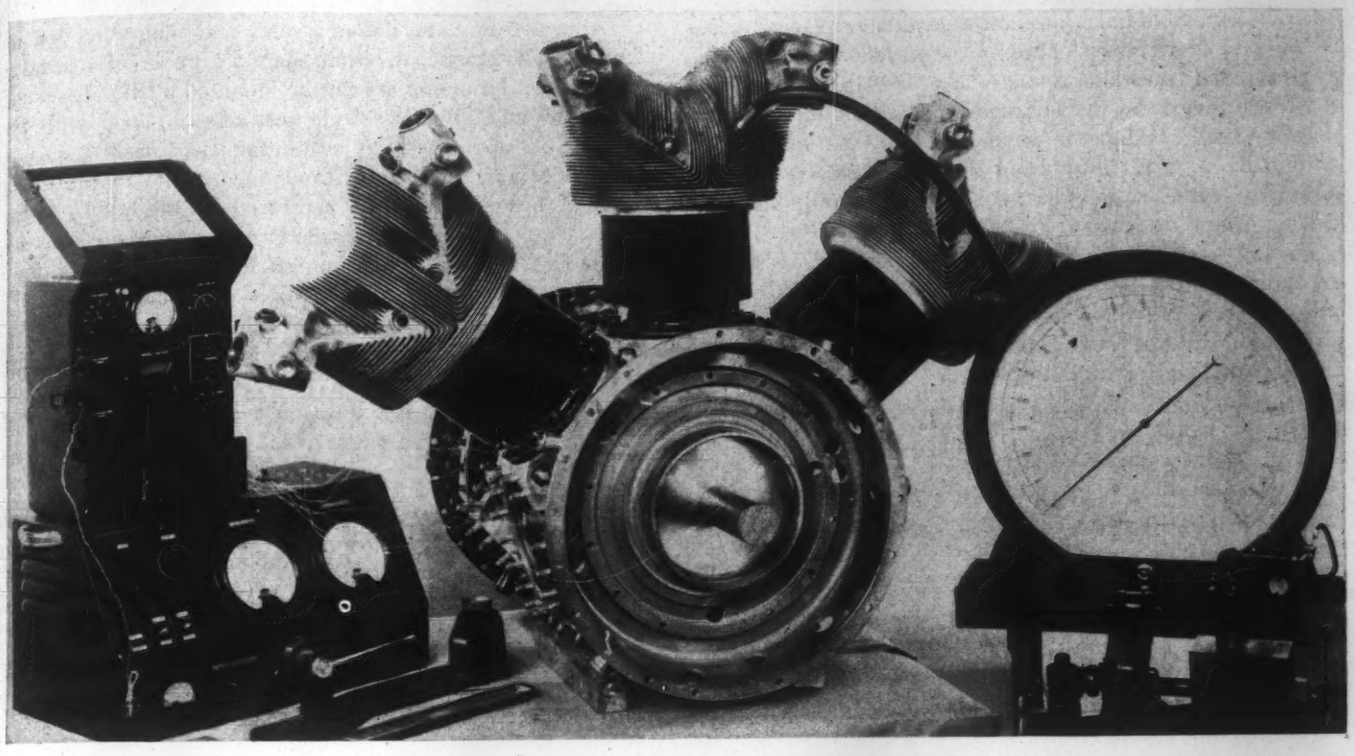

Improving Machine Parts Through Strain Measurement

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Cleveland

POSSIBILITIES of increasing life and output of machines as the result of stress analysis have captured the interest of engineers. Inability to estimate stress values or to predict the life of machine parts by mathematical calculations has stimulated the use of brittle lacquers, mechanical extensometers and electrical strain gages for the direct measurement of the strains which prevail under load at the surface of an actual machine part.

Fig. 1—Engine crankcase set up for strain analysis, using hydraulic pressure to simulate explosion pressure. Stresscoat and resistance-wire strain gages are used for strain measurements

PRACTICAL DETERMINATION of surface stresses by strain measurement can furnish the designer with much valuable information leading to improved life of highly stressed parts. In this article, based on a paper presented at the recent national aeronautic meeting of the S.A.E., the author gets down to cases and interprets the results of some actual strain measurements on machine parts. Also discussed are the benefits resulting from certain types of prestressing during fabrication and assembly

Discussion of methods for stress determination can be found in published literature (1,2,3)*. The purpose of this article is to interpret some of the results of such analyses.

Use of stresses calculated from assumed loads may be misleading as a criterion of the strength of a design. The inadequacy of this practice

* Numbers in parentheses refer to references at end of article.

arises from many factors difficult to evaluate. One of these is the fact that residual stresses or prestresses prevail in a great many machine parts when assembled, and the stress resulting from the imposed load is superimposed on them. These residual or prestresses may be set up intentionally or may result from the fabricating procedure and may cause a combined-stress condition difficult to calculate.

In determining the safe allowable imposed stress value, recognition must be given to the residual stress and its influence on the allowable imposed stress. The major stresses in an operating machine and their origins may be classified as follows:

1. Imposed stresses resulting from operating loads
2. Residual stresses arising on the release from nonuniform plastic deformation
3. Prestresses introduced during assembly
4. Thermal stresses set up during operation as a result of thermal gradients.

IMPOSED STRESSES: In *Fig.* 1 is shown the equipment used to load a German B. M. W. engine crankcase center section with a hydraulically simulated explosion pressure of 1500 psi in the combustion chamber of one cylinder. *Fig.* 2 shows the location and magnitude of the strains so developed. These strains have been evaluated by the use of Stresscoat and indicate the direction and amount of strain under the specific conditions.

Additional strains are generally present as a result of unbalanced crankshaft forces. These additional strains may increase or decrease the level of the strains resulting from explosion loads. Providing no change in the direction of principal strain takes place due to these additional strains, biaxial strain measurements can be obtained from two gages at each interesting location.

After locating the higher strains and their direction by the use of Stresscoat, strain gages capable of operation at elevated temperatures can be mounted at the interesting locations and dynamic strain readings obtained with the engine in operation. By mounting two strain gages at each location, one parallel and one normal to the direction of principal strain as indicated by the cracks in the Stresscoat, the strains may be used to compute stresses.

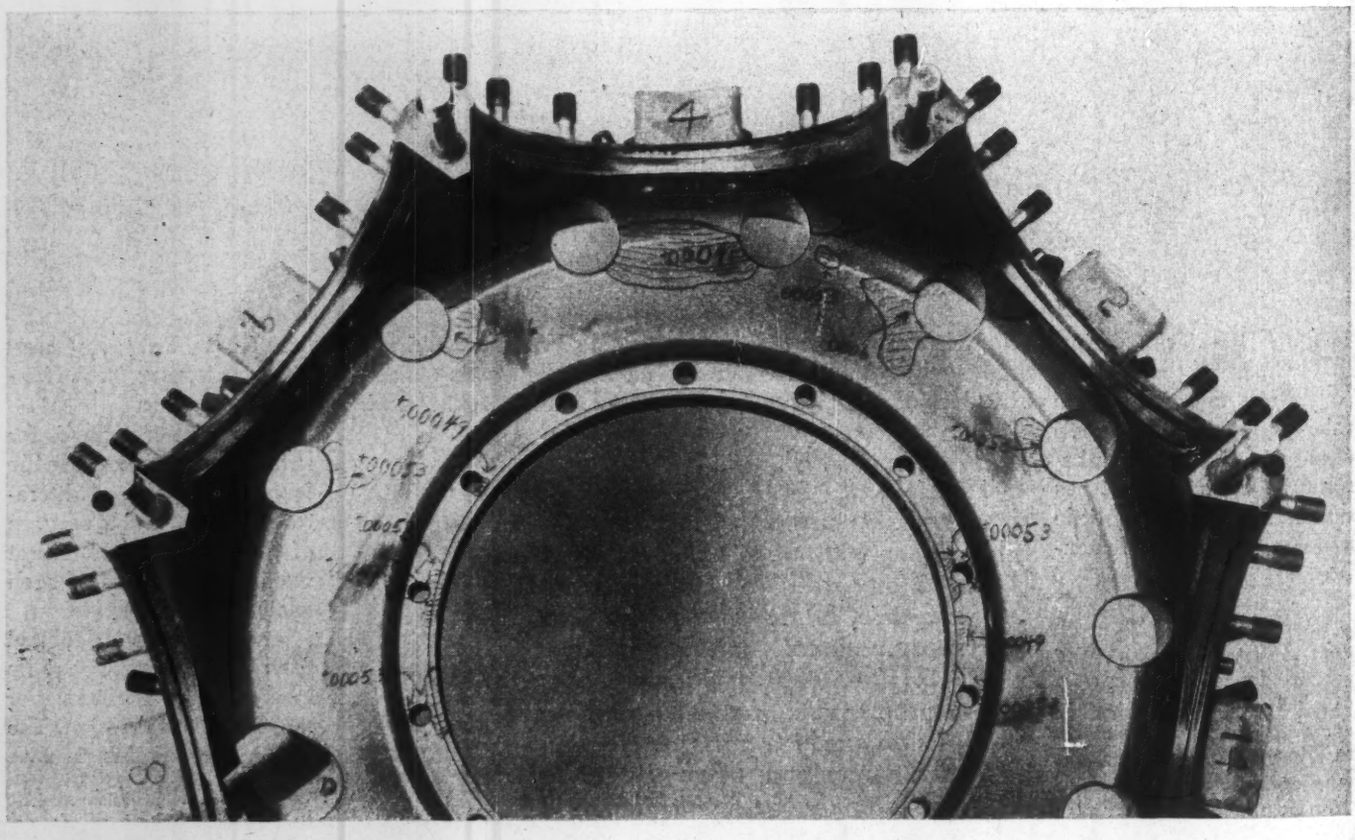
In *Fig.* 3 are shown formulas for computing the indicated stress resulting from biaxial strain. When the principal strain is multiplied by the modulus of elasticity of a material, the value obtained is expressed as apparent stress. A high degree of accuracy is obtained only when the principal strain is within the elastic range of the material, and when the strain results from pure tension or compression.

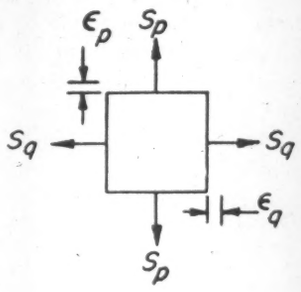
RESIDUAL STRESSES: Residual stress may be set up in a part during casting, fabrication, heat treatment, machining, or assembly by nonuniform cold plastic yielding of the material. With the exception of the X-ray method, residual stress determination is destructive to the part. The Sachs method (4) of residual stress determination is based on the fundamental phenomenon that the removal of part of a stressed body causes the remainder to suffer elastic strain.

Biaxial residual stress determinations at the surface of a part may be made by attaching wire strain gages at locations of interest and then carefully removing all the metal from the section except just that required for support of the gages (5). The approximate residual strains may be deduced from the readings of the gages.

A qualitative method of residual stress determination

Fig. 2—Stresscoat pattern outlined on inside surface of engine crankcase, showing magnitude of strain





$$\epsilon_p = \frac{S_p}{E} - \mu\frac{S_q}{E}$$

$$\epsilon_q = \frac{S_q}{E} - \mu\frac{S_p}{E}$$

$$S_p = \frac{E}{1-\mu^2}\left[\epsilon_p + \mu\epsilon_q\right]$$

$$S_q = \frac{E}{1-\mu^2}\left[\epsilon_q + \mu\epsilon_p\right]$$

where E = Modulus of Elasticity
μ = Poisson's Ratio

Fig. 3—Formulas for calculating indicated stress resulting from biaxial strain. Sketch gives notation

may be made by the use of Stresscoat. In *Fig.* 4 is shown a partial view of the head of a forged and heat-treated piston which had been coated with Stresscoat. Careful drilling with ⅛-inch sharp drill to a depth of approximately ⅛-inch caused a redistribution of stress to take place, thus causing the Stresscoat to crack. Cracks in the coating concentric to the drilled holes indicate that a state of compressive residual stress prevailed at the surface of the piston head. If the cracks had radiated from the drilled hole, a state of residual tensile stress would have been indicated. The random cracks in the Stresscoat between the drilled holes result from a balanced biaxial stress. For illustrative purposes, this piston had been heat treated in a manner which set up a relatively high level of residual stress in the piston head.

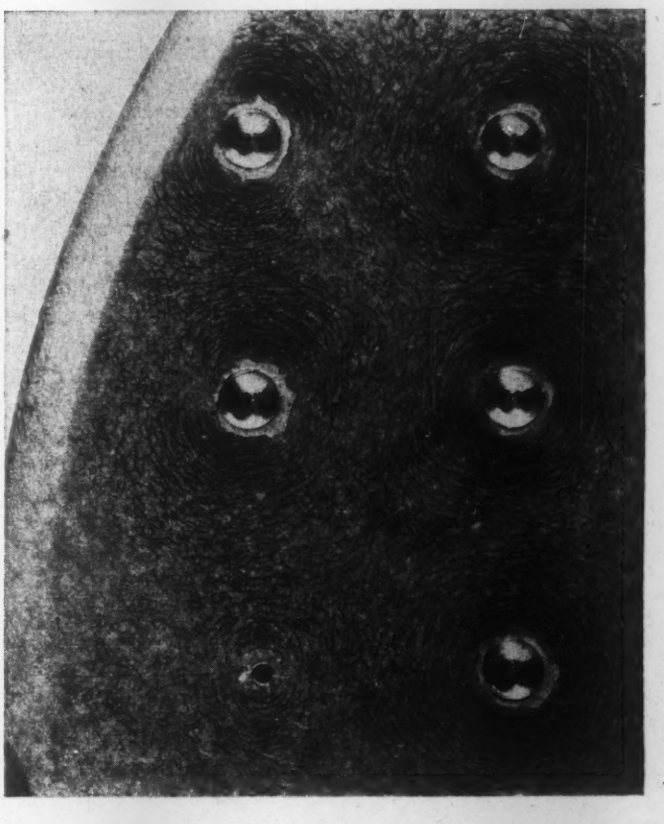
Quenching Stresses: Any nonuniform temperature distribution within a part induces stresses. The distribution and magnitude of the stresses may be affected if some portions yield during cooling. Residual stresses can be produced only if portions of the part are plastically deformed during cooling. The portion which cools most rapidly retains compression stresses, whereas the part which is last to reach room temperature contains residual tensile stresses. A favorable residual stress distribution for parts subject to repeated stressing often can be obtained by quenching the part in a predetermined manner.

Machining Stresses: Surface working produced by a sharp cutting tool, where the effect is limited to a few thousandths of an inch in depth, does not materially increase the fatigue strength of a part. Surface rolling in which an appreciable amount of plastic deformation has taken place in the surface layers may considerably improve the fatigue resistance. The usual distribution of residual stresses in a body rapidly cooled from an elevated temperature is such that the stresses at the surface of the body are compressive and, in the interior, tensile in nature. Removal of a portion of the body by machining may bring about a less desirable distribution of stresses. For example, the nature of the stresses in the surface layers may change from compressive to tensile.

Nonuniform Cold Reduction: Residual stresses resulting from plastic yielding during fabrication, assembly, or operation may be beneficial or detrimental to the fatigue life of machine parts. This depends on the relation of the residual stress direction to the direction of the imposed stress and upon the conditions which may be set up by the different stresses. Beneficial amounts and directions of prestress can be obtained by design and also in the fabrication of machine parts. Shot-peening (6), which sets up compressive stresses in the surface of a part, has proved its value in increasing the life of a part subjected to repeated tensile bending stresses. A beneficial effect can be obtained on machine parts by deforming the surface that is subjected to fatigue, by mechanical working. For example, the fatigue resistance of parts with transverse bores, keyways, screw threads, grooves, and fillets may be increased by plastically deforming the surface with suitable tools.

PRESTRESSING DURING ASSEMBLY: An increase in the maximum allowable operating speed of rotating parts often can be obtained by so-called "overspeeding". For simplicity, a uniformly thick circular disk with a central hole will be used as an example. It is assumed that the thickness of the disk is relatively small in relation to its diameter, and homogeneous material is used for its construction. Upon rotating this disk, tangential stresses are set up in

Fig. 4—Concentric cracking around drilled holes in a piston head, indicating that a compressive residual stress condition must have existed before drilling



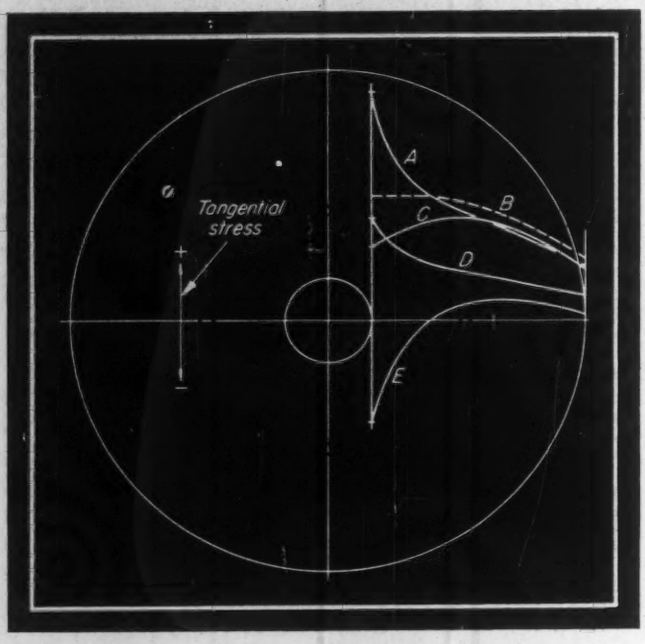


Fig. 5—Stress distribution curves indicating how beneficial prestresses are set up by overspeeding a rotating disk

the metal surrounding the central hole. Such a stress condition is shown in *Fig.* 5, the tangential elastic stress distribution, symmetrical about the axis of the disk, being illustrated by curve *D*. Upon increasing the rotative speed of the disk, a speed is reached at which the metal surrounding the central hole becomes stressed above the proportional limit but below the ultimate strength of the material. The theoretical stress distribution for this condition, assuming elastic action. is illustrated by curve *A*. This stressing is not compatible with the yield strength and a more nearly correct distribution is shown by the broken line, curve *B*.

Upon stopping the disk, a tangential residual stress condition will be set up, resulting from the permanent set of the material at the hole. The portion of metal which has yielded is compressed by the elastically strained outer portion and equilibrium is established by the tensile stresses set up in the outer portion, curve *E*. The tangential residual stress produced at the hole by stressing the material in the plastic range is opposite in sign to the stressing produced by rotation of the disk. This condition produces a favorable effect on the final distribution of the stresses in the disk.

When the disk is again rotated, the tangential stressing in the disk may be obtained by the algebraic addition of the elastic curve *A* and the residual curve *E*, shown by curve *B*. Stress distribution for any speed below the proof speed may be obtained in like manner.

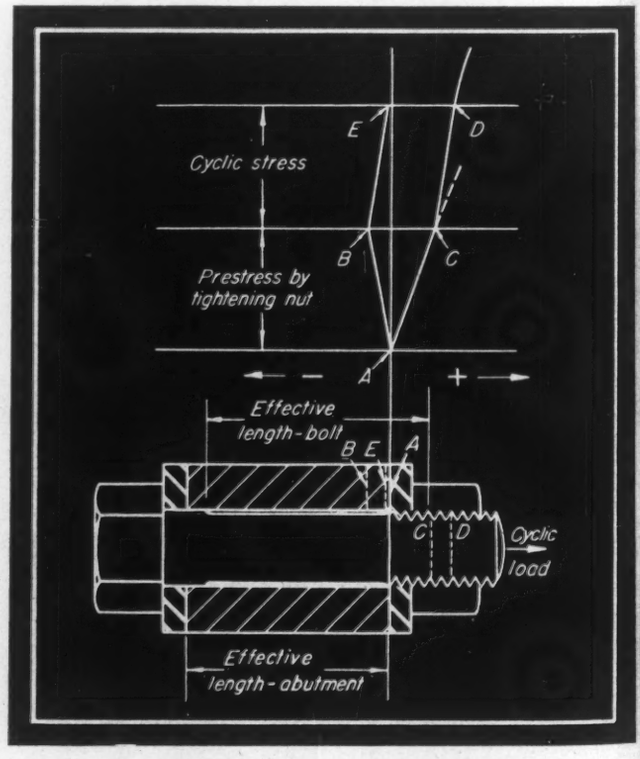
If the proof test speed is set to produce stressing as shown by curve *B*, the stress distribution at the rated speed will be represented by the curve *C*. The maximum tangential stress in curves *C* (with prestress) and *D* (without prestress) are equal. If curve *D* represents the tangential stress distribution at the maximum rated speed where the stresses are within the initial proportional limit of the material, the improvement resulting from overspeeding will be the difference in speed required to produce the stress distributions as shown by curves *D* and *C*.

Upon increasing the rotative speed of the disk above the previous maximum speed, yielding will again take place and a new level of residual stresses will be set up on returning to zero speed. No favorable effect will be obtained in overspeeding a rotative part unless the stress at some section of the part reaches a level above the proportional limit of the material.

Screw thread fastenings can be used to set up prestresses in assembly, whereby an improvement in life of the parts can be obtained. A bolt and a tube, *Fig.* 6, illustrate the phenomenon which takes place upon applying an additional load to a prestressed bolt. As the nut is tightened, the bolt will elongate from *A* to *C* according to the modulus of elasticity of the bolt material, the area of the cross-section, length of bolt, and load. At the same time, the tube surrounding the bolt will shorten from *A* to *B*. This shortening will be governed by its modulus of elasticity, cross-section area, and effective length, and is indicated in the stress-strain curve shown above the bolt illustration. When a load is now applied to the bolt as indicated by the arrow, it will elongate from *C* to *D* but, at the same time the tube will also elongate as shown by *B* to *E*. The new rate of deformation with the prestressed bolt will be governed by the ratio of the effective cross-section of the tube, effective cross-section of the bolt, their effective lengths, and their respective moduli of elasticity. Changing any of these design features will change the rate of strain in the bolt and tube when the assembly has been prestressed.

Generally, screw thread fastenings are prestressed in service. The amount of prestress should be governed by the service requirements. In *Fig.* 7 is shown the effect of prestress on the rate of strain for a bolt similar in design to *Fig.* 6. Data used in the construction of this diagram were obtained from a heat-treated steel bolt installed in

Fig. 6—How rate of deformation of prestressed bolt changes due to effect of abutment



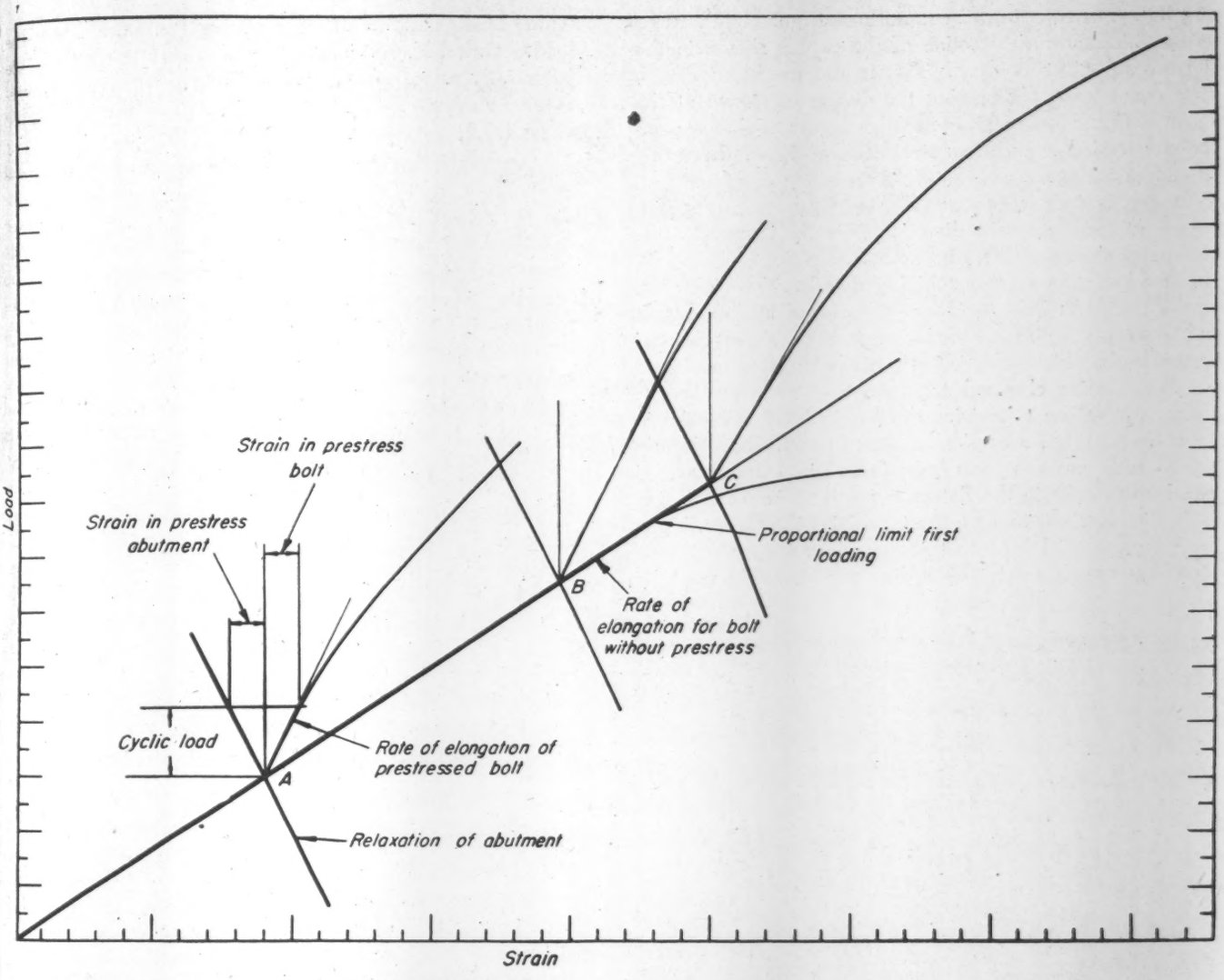


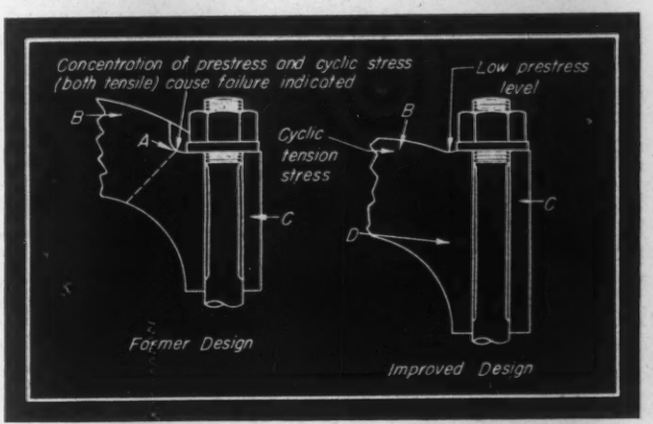
Fig. 7—Effect of prestress on rate of strain with different amounts of prestress, for steel bolt in aluminum boss

an aluminum boss. Baldwin-Southwark Type SR-4 strain gages of ¼-inch gage length were used for the strain measurements. The line connecting the points *A*, *B*, and *C* indicates the rate of strain when the load is applied to the bolt without prestressing. The proportional limit in tension is indicated for the first applied load on the bolt. The nut on the bolt was tightened, setting up a strain in the bolt to the level as shown by the point *A*. Upon applying external load to the bolt, the load-strain rate is less than the load-strain rate of the bolt without prestress. The rate of relaxation of the aluminum boss is shown intersecting the rate of strain in the bolt, at the prestress level. For a given cyclic load applied to the prestressed bolt, the cyclic strain in both the abutment or the bolt can be found as indicated in the figure.

At points *B* and *C* are shown the effects when greater amounts of prestress are used. When using the amount of prestress as indicated at *C*, the cyclic load which can be added without plastic yielding is of relatively small amount. By prestressing the bolt above the proportional limit of the material and then loosening, residual stresses will be set up in the bolt at the points of stress concentration. This residual stress will be beneficial when the bolt is subjected to fatigue. This is an example where improvement in fatigue life can be obtained by prestressing in the same direction as the imposed stress in the bolt and, by so doing, decreasing the stress range.

A type of design which has been the source of a considerable number of fatigue failures where screw thread fastenings have been used is shown in the illustration at the left in *Fig.* 8. This design has been used particularly in highly stressed connecting rod and main bearing caps. A spot face or counterbore provides a seat for the washer

Fig. 8—Typical design which may fail prematurely due to prestress and cyclic stress concentration at same location



and nut. The fillet in this counterbore is generally located at the minimum cross-section of the cap as shown in the illustration at the point *A*. This is also the location of a high cyclic stress. Changing the design as shown at the right in *Fig.* 8 lowers the tensile prestress level, the combined stress, and improves stress conditions of the screw thread fastenings.

In *Fig.* 9 is shown a method whereby prestressing may be set up in the opposite direction from the imposed stress to improve service life. In both illustrations, *A* and *B*, the left half shows the machining details. The right half shows the parts held together with one of the bolts used in clamping the parts. Illustration *A* shows a conventional assembly in which the mating parts have been machined parallel. When clamped together, any prestress in the radius where the tube joins the flange is of a low level. In *B* the mating surfaces have been dished. Consequently when these surfaces have been forced into intimate contact by the bolts in the flange, a prestress in compression exsists in the radius where the tube joins the flange. Upon applying an axial load to this assembly, before a tensile stress can be set up in this radius, the assembly must be

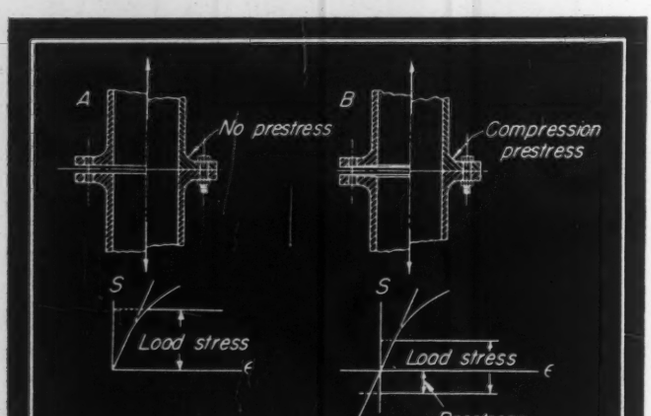


Fig. 9—Beneficial effects of prestressing an axially loaded member. Right-hand design employs dished flanges

loaded a sufficient amount to reduce the compressive stress to zero. This is shown in the stress-strain diagrams accompaying the illustrations. Strain gages applied at the radius in both designs would show strain readings approximately equal for a given load. However, the strain readings obtained from assembly *A* would represent tensile stress, while those obtained from assembly *B* would represent a strain range resulting from some value of compressive prestress. To evaluate the tensile stress, it would be necessary to determine the strain in compression set up at gage location due to the tightening of the flange bolts, and then subtract this value from the value obtained under load in order to determine tensile strain.

THERMAL STRESSES: Thermal stresses are induced by restraining the normal expansion or contraction which would accompany a change in temperature. A highly complex system of stresses may be set up within a part by the temperature gradients which generally prevail at elevated temperatures.

COMBINED STRESS: Residual, thermal, machining, and assembly stresses generally function as a prestress upon which is applied the stress resulting from the operating load. However, resultant stresses cannot be determined vectorially as is possible when dealing with forces, and careful analysis is necessary to determine the prevailing stress level.

ALLOWABLE STRESSES: For static loading it is relatively simple to determine the safe allowable working stress. On the other hand, fatigue strength values as determined by the usual laboratory tests cannot be used direct because such tests generally are performed on specimens carefully prepared to reduce to a minimum the effect of mechanical stress raisers such as grooves and scratches. Possibilities of designing a machine member free of stress concentration are extremely limited. Machine parts generally will show a resistance to fatigue somewhere between the stress causing failure in a smooth-finish test bar and in a notch fatigue test bar. The design may approach perfection but it will still be necessary to allow for the variation of properties resulting from manufacturing tolerances.

Mechanical properties of test specimens taken from machine parts generally are below those obtained from laboratory test bars of the same alloy and heat treatment. Examples of factors causing this difference are grain flow in forgings and rate of solidification in castings. In addition, there is the influence of surface irregularities, internal discontinuities, and abrupt changes in section—all of which greatly influence the strength of the part.

Material Often Inefficiently Utilized

SUMMARY: A few of many methods which can be employed to improve life of machine parts have been presented. Many failures of machine parts in service which have been attributed to material should have been charged to design. Machine parts generally contain a sufficient amount of material, although often the material is inefficiently distributed.

The criterion of the strength of a design should not be the magnitude of the stress calculated from the anticipated imposed loads in relation to any one property or characteristic of the material used. For economical reasons, designs should be proportioned to take advantage of any prestress which can be beneficial. Recognition must be given to the properties of the material in the actual finished piece and these weighed against the service requirements.

Direct measurement of strain requires considerable research and study before its advantages can be evaluated. It has already proven its worth in strengthening many machine parts. It points the way to the economical distribution of material in the part, thus improving the design and contributing to progress. It seems certain that the field of usefulness of analysis by direct measurement of strain will be greatly enlarged in the near future.

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Centrifuges

. . . with redesigned rotors yield more complete separation of two liquids

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DESIGN of centrifugals for fluid separation presents problems of considerable complexity, for the liquids may exist in widely varying proportions and their density ratios may approach unity. Further, most liquid combinations tend to emulsify to some degree, particularly under the highly turbulent conditions existing in modern high-capacity units.

A typical centrifugal problem is met in the purifying of transformer oil, performed in a machine such as shown in *Fig.* 1. In this instance, the oil must be so cleaned that all but a minute portion of water is removed. Illustrated in *Fig.* 2 is a type of bowl developed for this service. Under centrifugal force, the liquid is separated into two layers, the heavier moving to the outside and the lighter forming the inner cylinder. An annular disk, or dam, is located near the bowl top so that the lighter liquid may overflow the inner diameter and then be delivered to the outside through drilled holes. The heavier liquid escapes past the outside diameter and is then led inward, overflowing the inside diameter of a replaceable ring. Balance of layers is created by a U-tube effect, the pressure of the combined light and heavy layers beneath the inside disk equalling the pressure of the heavy layer above the disk. The dam over which the heavy liquid discharges is supplied in various diameters, and a selection is made to position the dividing line between the two liquids at the most advantageous place. For a given position of dividing line, widely different heavy dam diameters are required for various differences of specific gravity. By making the dams over which the two liquids discharge completely annular, cresting over the dams is minimized, thus permitting wide variations of rate of feed without materially altering the position of the dividing line.

There are not many mixtures of immiscible liquids that do not tend to emulsify to some extent. As a result there is, adjacent to the dividing line between the light and heavy phases, a layer of emulsion which may be in the water layer or in the oil layer depending on the type of emulsion formed (that is, whether the emulsion is a continuous phase of water with droplets of oil suspended in it, or a continuous phase of oil with droplets of water in suspension). If the distance between the light and the heavy draw-off point were wider than the normal emulsion layer it would be possible to get clean discharges from both sides

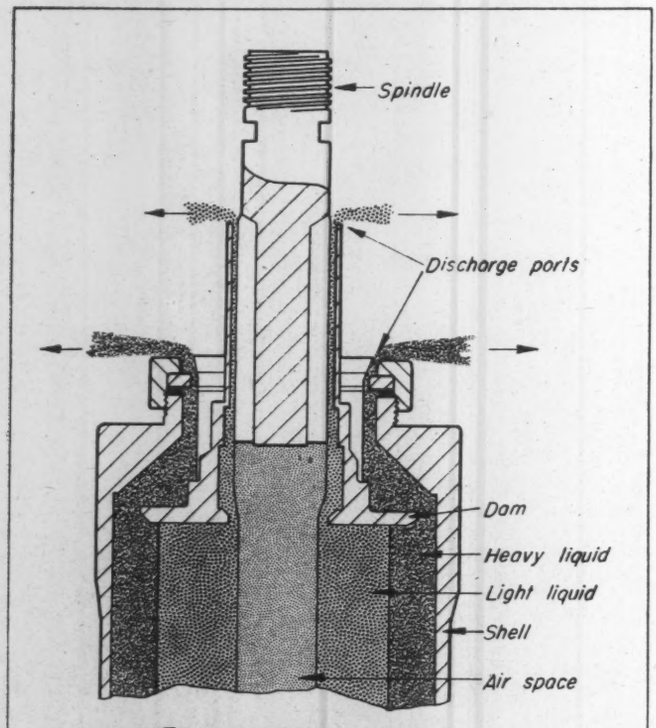
Fig. 1—Centrifugal separator incorporates all controls and accessory equipment within the machine structure. Control center is rectangular box to right of the centrifuge

provided the overflow over the light dam was directly from the surface. However, it is a characteristic of such liquid flow that a portion is drawn from beneath the surface and it is, therefore, dangerous to permit the emulsion layer to encroach far toward the center.

A striking improvement in centrifugal bowl design is illustrated in *Fig.* 3. By enlarging the bowl top it was possible to make the outside diameter of the top disk larger than the diameter of the bowl proper. With this arrangement, the dividing line can be located so that there is only light liquid in the bowl proper. This means an unusually deep layer of such liquid, resulting in its being under the influence of centrifugal force for a maximum time. The heavy liquid layer is removed from the light-dam diameter so that even when a portion of this overflow is from beneath the surface the light discharge is not contaminated.

As capacities increased a point was reached where the light-discharge holes were not capable of handling the flow without creating back pressure that radically shifted the dividing line, thus upsetting separation. Lacking space for larger holes, the problem was successfully solved by a redesign that tilted the discharge holes outward, utilizing the tremendous centrifugal force available to empty these holes. This arrangement is shown in *Fig.* 4.

Fig. 2—Simple bowl design. Annulus near top of bowl space forms dam to control rate of effluent discharge



Liquid to be separated is introduced into the rotor by means of a jet at the bottom. Interior of the bowl is equipped with a wing device to bring the liquid up to the speed of bowl rotation, to accelerate toward the outside any of the heavy phase and to decelerate toward the inside any of the light phase. If the liquid is jetted directly against the wing device it is sheared abruptly. This may create emulsions that are hard to break, presenting additional problems.

It has been found that the most satisfactory method of introducing liquid into a rotor of this kind is to equip the lower end of the wing device with a cone, the maximum diameter of which is greater than the air core. The incoming jet strikes the point of the cone and the liquid is gradually brought up to speed through slippage against the cone and against the body of liquid in the bowl. Provision for the outside diameter of the cone being greater than the air core accomplishes two purposes: One is to make sure that the jetted liquid does not pass the cone and strike the rotating vanes, and the second is to prevent air communication up the center of the bowl. This might carry a portion of the incoming feed direct to the light-discharge holes rather than permit it to enter the subsidence section where it can displace separated liquid at the top.

Vapor Contamination Problem

One important problem was to devise means of delivering the separated effluents from the machine without the possibility of remixing through intercommunicating air currents. Normal shape of a bowl-top is such that air is nomally drawn in at the top, at the point of smallest diameter, and blown out at the larger diameters, thus tending to cause vapor contamination from the upper to the lower collecting cover. Various means have been adopted for solving this problem. One is to insert wings under the upper cover deck in such relation to the rotation of the bowl that the blowing is directed upward, in opposition to the normal air currents. This can be made highly effective for any given ratio of discharges, but is not of universal applicability since a certain amount of air is entrained with the discharge liquids. This in itself causes a variation in the direction of air flow, depending on the proportion of the discharges. The best answer is to interrupt the free flow of air currents by shutting off all points of air ingress and egress. This not only tends to solve the cover leakage problem but also avoids delivering oil or other vapors into the room through the discharge spouts.

Shown in *Fig.* 5 is a general arrangement of such a vapor-tight machine. Mounted in the frame, to receive the two liquid discharges from the bowl, is a casting containing two concentric channels each of which is connected to the outside through piping. The lower cover is a conical disk of cast stainless-steel. The upper cover is a conical disk of drawn stainless-steel sheet, and the cover lid is dome-shaped, of drawn stainless-steel sheet. Between

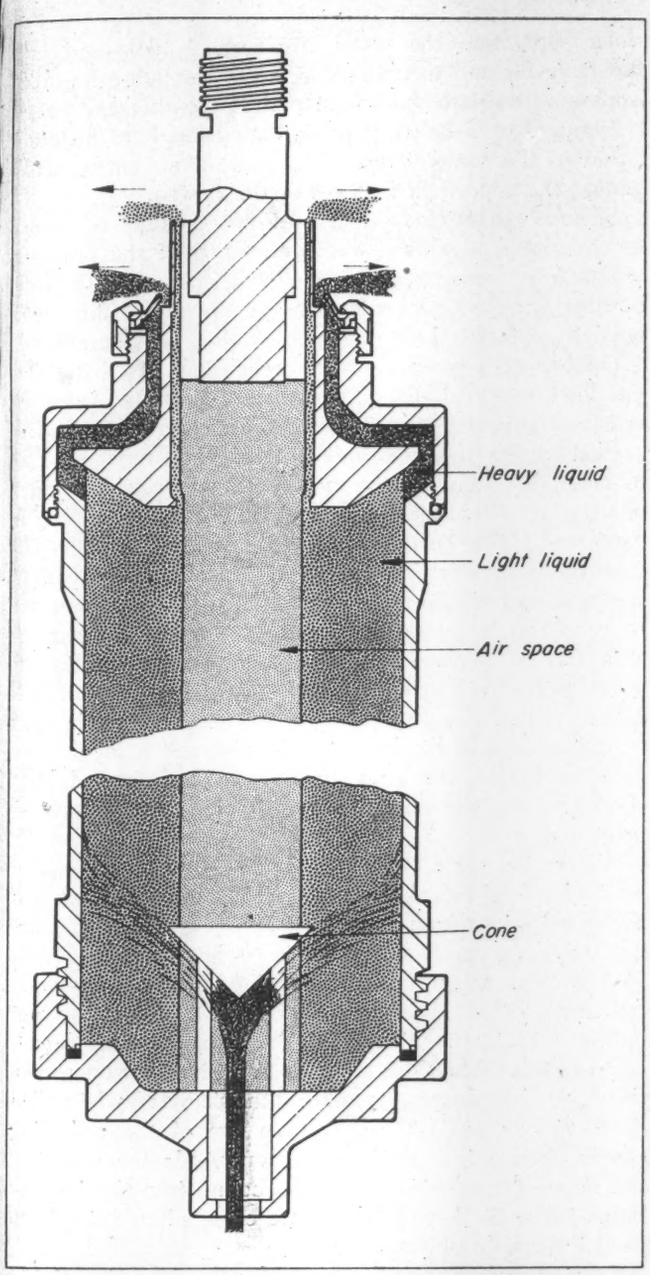


Fig. 3—Modified bowl design. Increasing diameter of the annulus increases the liquid radius, increasing migration time of the heavy particles. Improved separation results

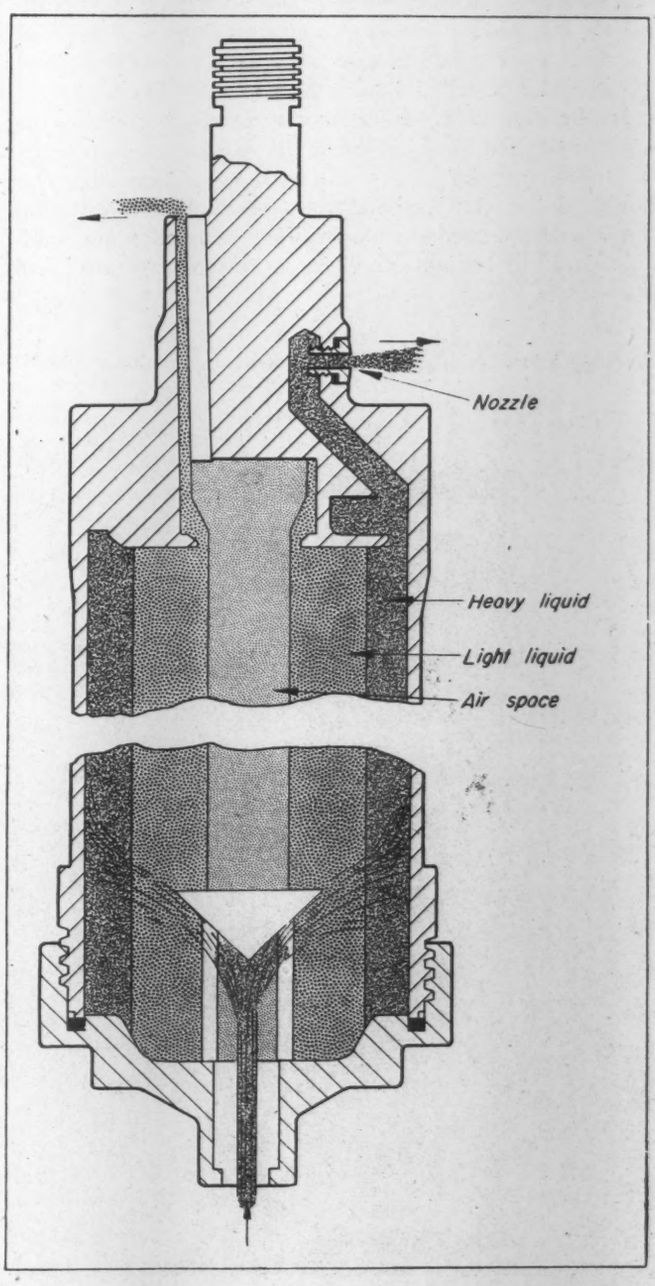
the cover lid and the cast cover base is a gasket which is sealed tightly by pressure from the spindle guard. A gasket is also provided beneath the upper cover and, to guard against leakage of the liquids from one compartment to the other, arrangement is made to pull this cover down tightly by means of swing-bolts equipped with wing nuts. The slotted pieces, against which the wing-nuts face, are formed of heavy stainless-sheet and spot welded to the cover deck.

Light liquid is delivered from the cover base casting through a short length of tubing which enters a funnel, not shown in *Fig.* 5. This funnel is equipped with a baffle to maintain a depth of liquid in the trapped portion, and the tubing from the cover base is extended so that the liquid is delivered beneath the surface of the body of liquid in the funnel trap. By such means the outlet on the light side is completely sealed against air ingress or egress, preventing a circulation of air currents that might result in leakage from one cover to another and also preventing the discharge of objectionable vapors into the room.

When such a machine is in operation, liquid discharging into the upper cover entrains air that is originally there and additional air that tends to leak past the gasketed joints, thus maintaining a partial vacuum in the upper chamber. A collar is extended below the deck of the upper cover so that there may not be creepage of liquid across that surface and into the upper compartment.

Suspended by means of a flexible spindle from a ball-bearing assembly, the bowl is guided at the bottom by a bushing so mounted that it may shift to permit the center of rotation to be the center of mass, an essential requirement when the unit is to be operated at above critical speed. Drive is through a fabric belt from a vertically

Fig. 4—Improved discharge method. Radial discharge hole augments hydrostatic head increasing effluent rate

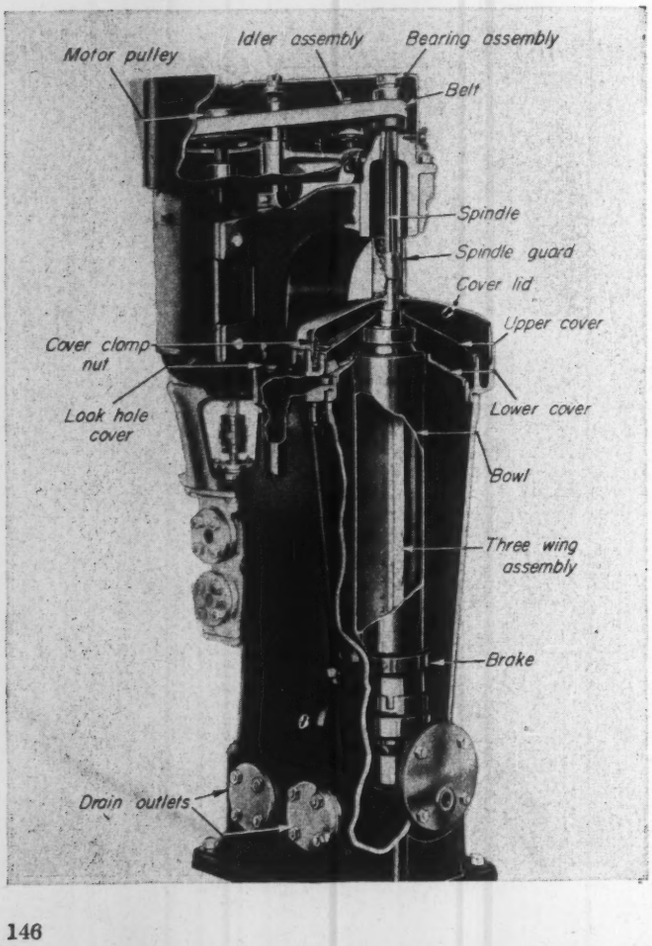


mounted motor on the back of the frame. Paper pulleys formerly used suffered from a separation of laminations in humid atmospheres. These have now been replaced by ones molded from Bakelite. Since this material is intrinsically low in coefficient of friction much experimentation was required to get a filler that would provide proper traction. This pulley has a slight crown for proper tracking. Very little crowning, however, is required at the belt speed of 9000 fpm utilized. High crowns must be avoided as they tend to split the belt.

Belt tension is maintained by an idler pulley whose arm is controlled by a torsion spring. This is most satisfactory, keeping a constant pressure on the belt and preventing bouncing that is often incident to the use of a counterweighted idler. The motor has a double-shaft extension and a pair of tandem-mounted pumps are coupled to the lower shaft end. This provides means for feeding liquid to the bowl and for delivering purified liquid from the machine. Since the motor operates at 3600 rpm, a screw type pump has been selected to minimize the reduction in suction lift from pump wear. Such a pump is particularly adapted to this service since it is able to feed an oil and water mixture without creating emulsions that would make centrifugal separation more difficult.

In the base of the frame is a reservoir into which the liquid from the bowl drains at the end of the run, when the machine is shut down. Piping is so arranged that this liquid can be fed to the bowl when the machine is re-started thus creating a needed water seal and reclaiming any valuable liquid that might otherwise go to waste. For oil and water separation the water discharge is delivered into this reservoir and then flows to waste through a line that communicates with the lowest part of the chamber. Then it is raised to a delivery point high enough to maintain liquid in the reservoir and thus prevent air entry, maintaining the vaportight features of the design.

Fig. 5—Cutaway of centrifugal showing machine details



Smooth, easily-cleaned frames are created by cored sections, and advantage has been taken of the presence of a hollow interior to mount the motor starters, with push-buttons arranged convenient to the operator. Since the bowl is an inertia load, the motor draws high current for a number of seconds during acceleration. To withstand this load, overload-heaters would need to be too large for satisfactory protection. The standard control adopted is a combination of two switches of magnetic type, the first to close the circuit to the motor and the second to short out the overload devices as long as the "start" button is depressed. The operator holds this button in while the machine is accelerating. After the machine has reached speed, release of the button permits the shorting contactor to drop out, allowing the overload heaters to function. With this arrangement it is possible to use overload heaters of the exact load rating. In series with the stop button is a limit switch mounted adjacent to the idler arm in such a position that if the belt stretches beyond a satisfactory limit, or breaks, the limit switch will be depressed, thus shutting down the centrifuge and pumps for belt replacement.

All moving parts are enclosed. The belt is surrounded by a completely shielding belt-guard, of sheet metal, and the spindle is surrounded by a spindle guard which acts not only as a protection for the operator but also seals against air ingress into the discharging liquid, as previously described. This sleeve has an external thread at top and bottom, with intermediate relief. The bearing base is threaded to match. When this sleeve is in position for operation, the upper threads are engaged and the sleeve may be pulled down tightly against a gasket on top of the collecting covers. When it is desired to disassemble the machine, the sleeve is unscrewed and pushed up into the bearing base. Then the lower threads are engaged sufficiently to keep it from dropping down.

In many cases it is necessary for the incoming liquid to be heated. For such applications the centrifuge is mounted on a sub-base in which electrical heaters are installed as shown in *Fig.* 1. A removable plate gives access to wiring space, and allows easy removal of the heater elements. A similar plate at the rear of the unit permits straight-through cleaning of the tubes in which the elements are placed. This is a necessity when handling material such as diesel lubricating-oil, since carbon tends to bake onto the elements, reduces heat transfer and may eventually result in heater burn-out.

A heater contactor is mounted in an enclosure beside the centrifuge, and the face of that enclosure is utilized as a compact control center, including a valve to regulate the rate of feed, and adjustable thermostat to maintain a constant desired temperature on the incoming oil, a dial type thermometer and a matching dial pressure gage to indicate feed rates. The unit is, therefore, self-contained, providing all that is necessary to reclaim lubricating or fuel oil and requiring only the connection of the feed wires and piping from used oil supply to the unit and return piping to deliver oil to the system or to storage.

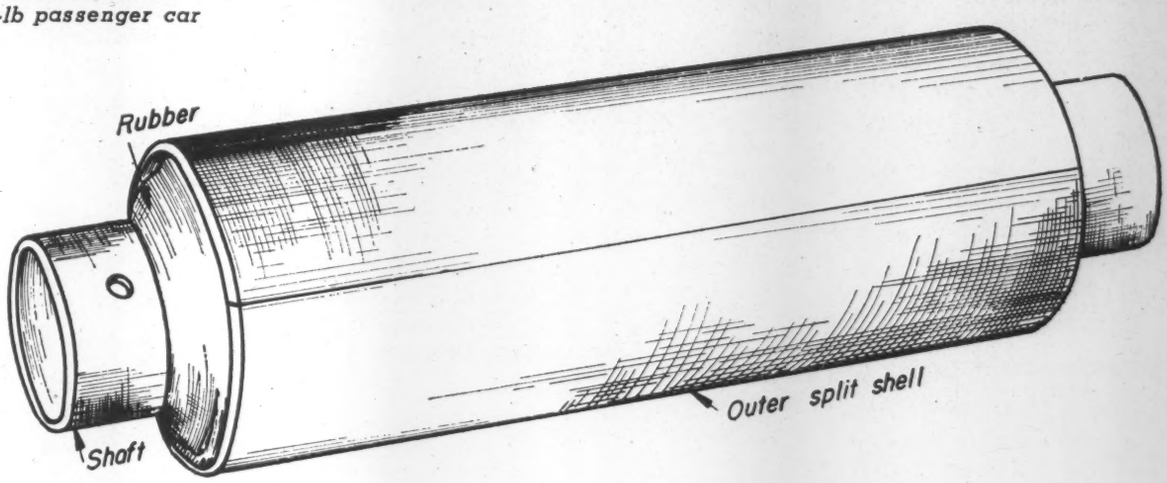


Fig. 1 — Torsilastic spring weighing 10.74 lb., used on the front-end suspension of a 3400-lb passenger car

How to Design Rubber Torsion Springs

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RUBBER and synthetic rubber are ideal materials for many types of mountings and suspensions. Possible applications cover a wide range of designs —from a simple rubber pad under a motor to a highly developed spring suspension for a passenger car. Of these, the torsion spring presents distinct advantages not only in its characteristics as a spring but also in flexibility of application, which makes it possible to meet a wide range of requirements by variations in spring design and in the length of the moment arm which applies the torsional load.

The Torsilastic spring is a torsion type spring having an inner shaft surrounded by an annular layer of rubber bonded to it and to a metal shell around the outside, *Fig.* 1. Load is usually applied at the end of an arm attached to the inner shaft, the outer shell being secured to a housing.

In considering the application of rubber torsion springs, the logical place to start is with the limitations of the application. From the proposed layout the length of effective arm can be determined, although rubber torsion springs permit somewhat more freedom since there are no coil

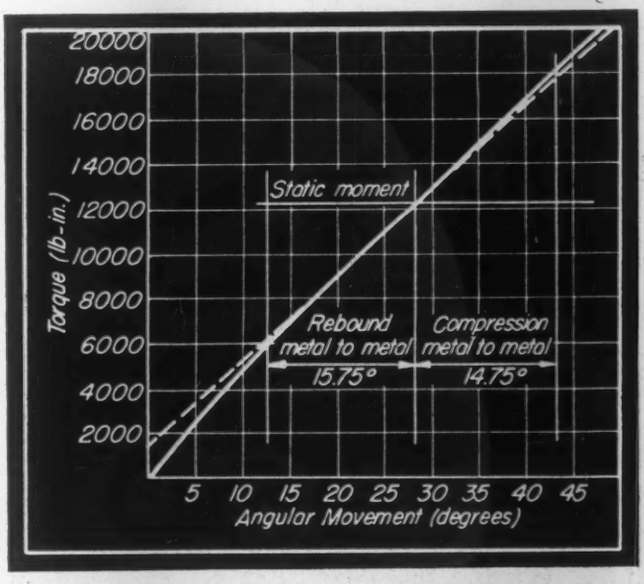


Fig. 2—Torsional loading curve for Torsilastic spring. Spring rate is virtually constant throughout useful range

spring seats, seat angularity problems, or high leaf spring stresses to limit the designer. With arm length and static spring load and static deflection determined the loaded moment and torsional rate of the spring are fixed, after which it is possible to proceed with problems of spring design.

A wide range of designs will meet any fixed requirement for static load and rate. That is, the spring may be long and slender or short and thick, depending on such considerations as clearances and the amount of flexibility desired in the mounting. It is, therefore, necessary to know something about the characteristics of such a spring before proceeding to more detailed spring design. The Torsilastic spring shown in *Fig.* 1 is typical of a design for the front suspension of a 3400-lb car and has the characteristics shown in *Figs.* 2 and 3.

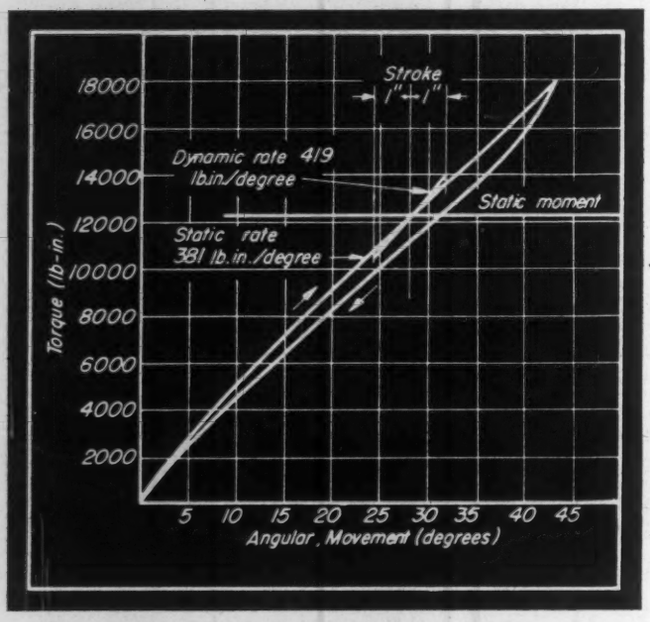
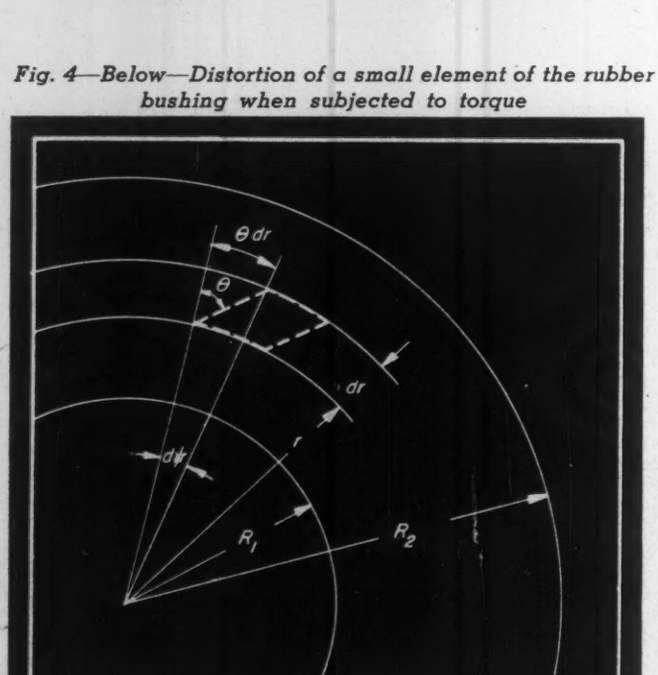


Fig. 3—Above—Comparison of static and dynamic rates for rubber torsion spring. Spread between lines for static cycle represents hysteresis or energy absorption

Fig. 4—Below—Distortion of a small element of the rubber bushing when subjected to torque



Rate or frequency of the cycle modifies not only the hysteresis loss but also the slope of the stress-strain curve for any rubber spring, *Fig.* 3. The dynamic rate is determined by an oscillation test made in the laboratory by suspending a mass on a moment arm which produces the static loaded torque on the spring. The mass is then excited and the free or resonanat period of oscillation observed. Length of the laboratory arm need not be exactly the same as the support arm to be used on the application since frequencies are easily converted into dynamic torsional rates by applying the familiar formula, $d = (188/f)^2$, where d is the deflection in inches and f is the frequency in cycles per minute. The deflection d is then the nominal length of arc through which the spring turns before it supports the average moment, and unit torsional rate is obtained by dividing the moment by the subtended angle.

Damping Negligible at Short Strokes

Under these conditions the hysteresis loss during the two-inch dynamic stroke is so small as to be difficult to show on a graph such as *Fig.* 3. This curve, while applying directly to Torsilastic springs, is similar to the curves for many other types of rubber springs and indicates the difficulty of dispensing with shock absorbers as is sometimes proposed where rubber springs are used. For large strokes the damping effect of hysteresis becomes noticeable, but for short strokes shock absorbers are desirable in passenger car suspensions, at least, to attain the standard of ride expected by the buying public in this country.

One other characteristic of these rubber springs, which is not easily shown in graph form, is the life of the unit. In attempting to forecast actual service life of a rubber spring by means of accelerated fatigue tests, it is necessary first to emphasize the exteremely wide difference between steel and rubber as elastic materials. Any acceleration in the normal flow of energy through a spring may reasonably be expected to have an exaggerated effect on rubber as compared with steel. This is substantiated by all available test data.

In any of the designs thus far considered there is very little exposed surface of rubber and any deterioration at these exposed ends has been found to have a negligible effect on spring life. It has not been found necessary to use a shield of any kind at the ends, and the aging of the body of rubber is extremely slow.

With some understanding of the proposed use for these springs, and of their characteristics, more detailed problems of spring design can be considered. With the required static moment and elastic torsional rate determined, the remaining design steps can be carried out very simply. The analysis for finding the dimensions of the rubber bushing in terms of the relation between torque, T, and angular deformation, ψ, (spring rate) is as follows.

In *Fig.* 4 is shown a small element of the material which is distorted through an angle $\theta = S/G$, where S is the unit shearing stress and G is the shear modulus. Distortion of this element contributes an amount $d\psi$ to the total angular deformation or wind-up, ψ. From *Fig.* 4,

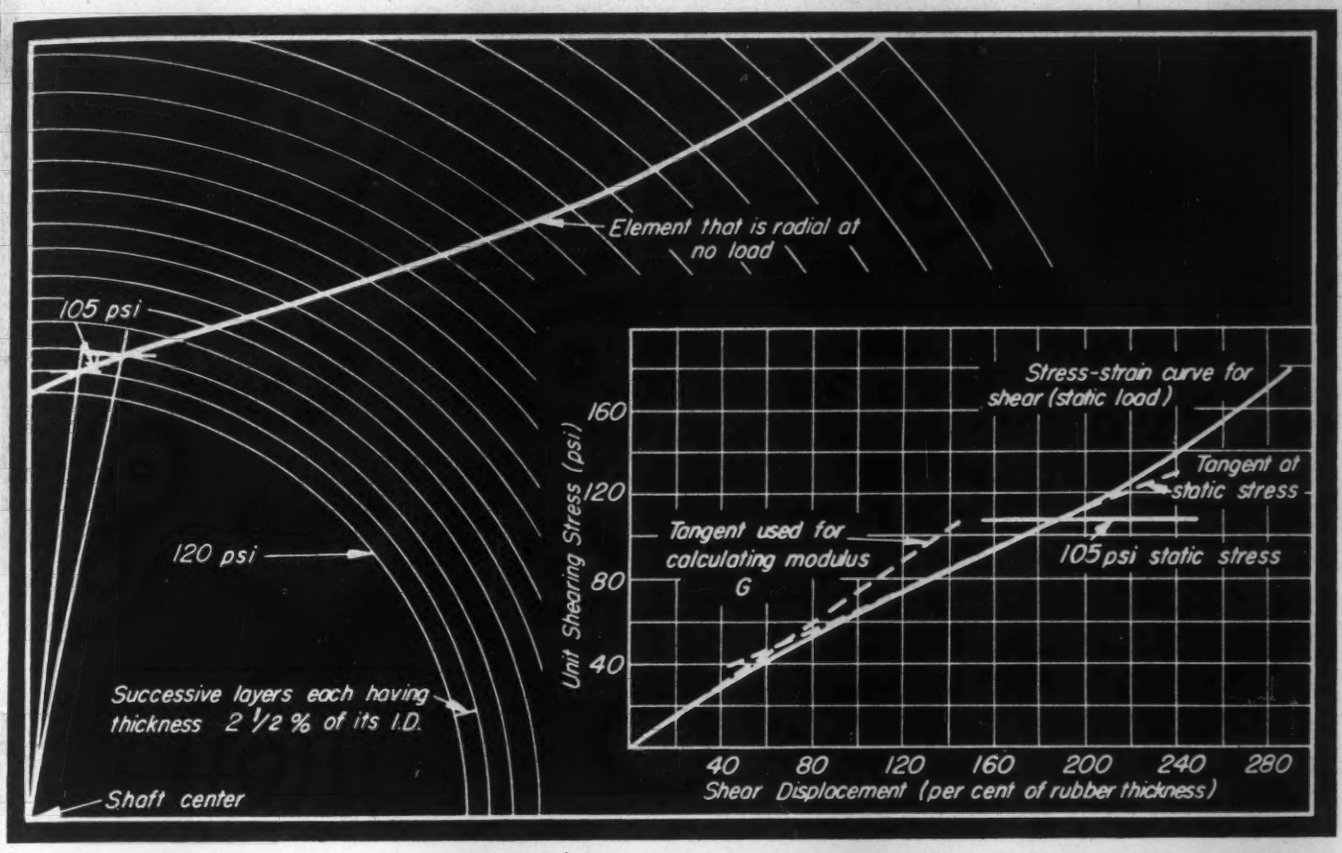

Element that is radial at no load
105 psi
120 psi
Successive layers each having thickness 2 1/2 % of its I.D.
Shaft center
Stress-strain curve for shear (static load)
Tangent at static stress
Tangent used for calculating modulus G
105 psi static stress
Unit Shearing Stress (psi)
160
120
80
40
40 80 120 160 200 240 280
Shear Displacement (per cent of rubber thickness)

Fig. 5 — Above — Section through a simple rubber torsion spring. Insert shows a stress-strain curve for a sample of the same rubber in "pure" shear as obtained on a shear sandwich

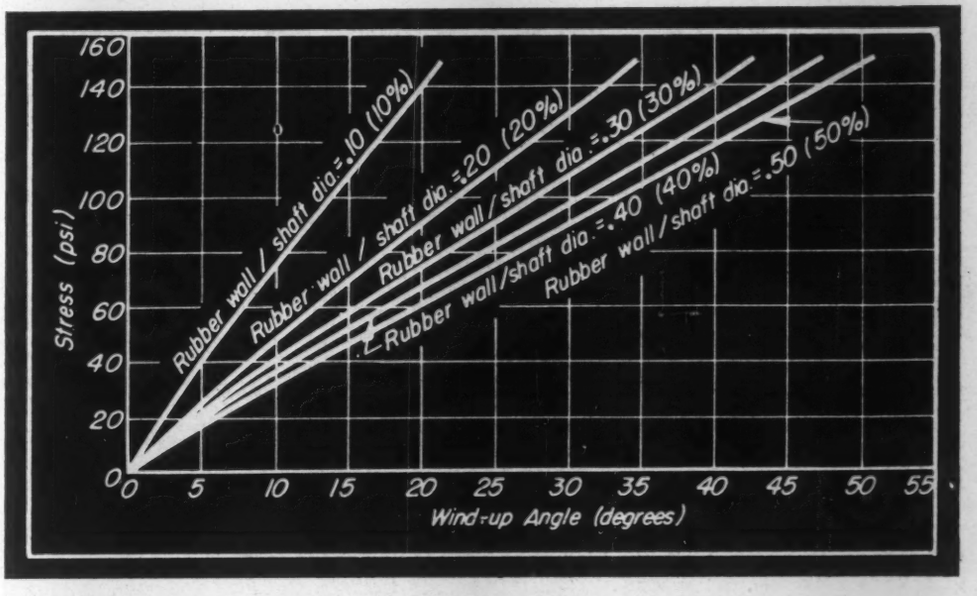


Fig. 6—Right—Rubber torsion spring characteristics under static conditions (slow wind-up). Dynamic rate (less than one cycle per second) is 25 per cent stiffer for natural rubber of 45 durometer, 50 per cent stiffer for synthetic rubber of 50 durometer

$$d\psi = \frac{\theta dr}{r} = \frac{S}{G}\frac{dr}{r} \quad \ldots\ldots (1)$$

Also, the torque is equal to the product of the cylindrical area of radius r and length L (the length of the bushing) times the shearing stress times the radius r:

$$T = 2\pi r L \times S \times r = 2\pi r^2 SL \quad \ldots\ldots (2)$$

But the torque is the same at any radius, therefore r^2S is a constant and may be set equal to $R_1^2S_1$ where S_1 is the stress at radius R_1. Then $S = R_1^2S_1/r^2$ and Equation 1 may be written

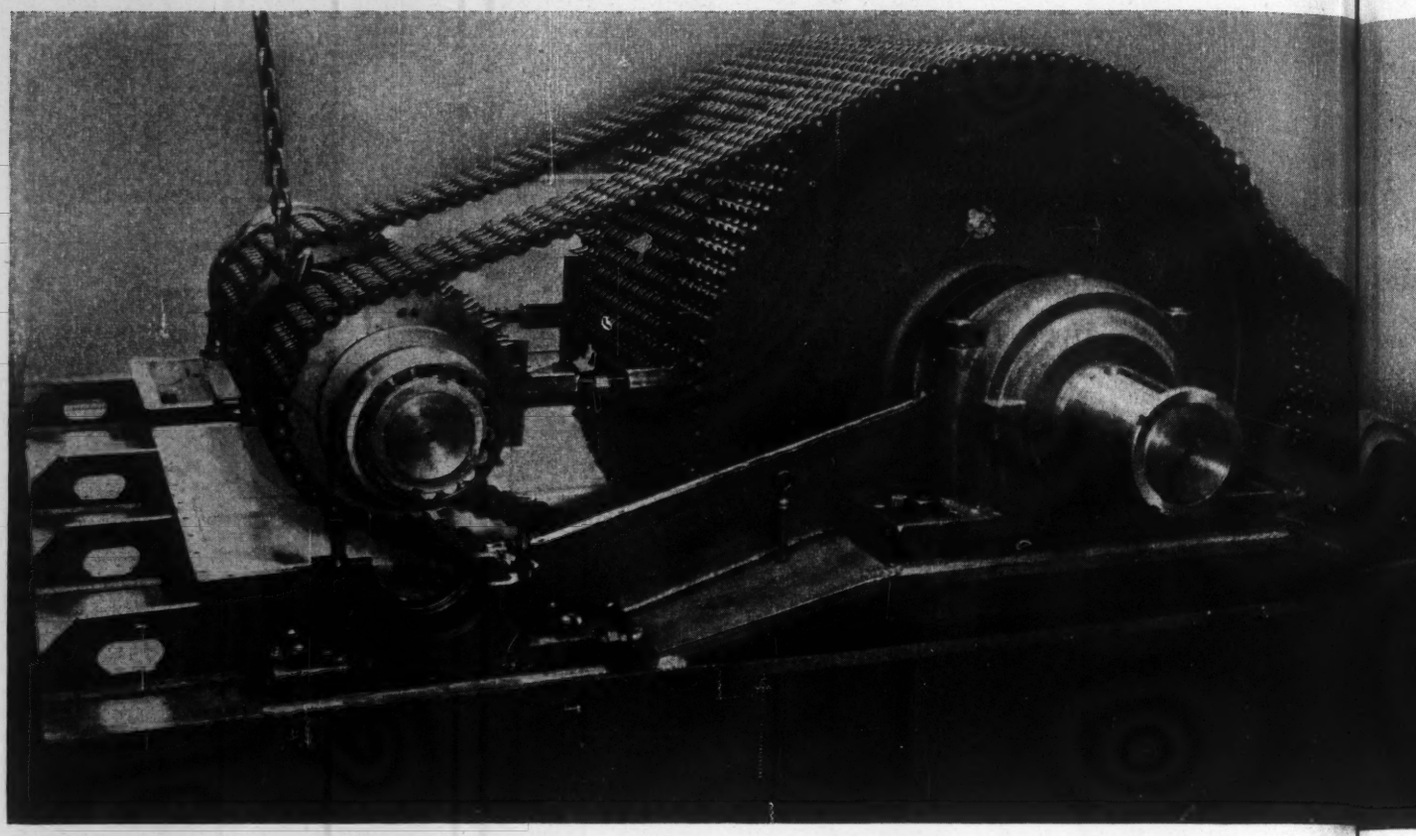
$$d\psi = \frac{R_1^2 S_1 dr}{r^2 Gr} = \frac{R_1^2 S_1 dr}{Gr^3} \quad \ldots\ldots (3)$$

Integrating between the inside radius R_1 and outside radius R_2

$$\psi = \frac{R_1^2 S_1}{G}\int_{R_1}^{R_2}\frac{dr}{r^3}$$

$$\psi = \frac{R_1^2 S_1}{2G}\left(\frac{1}{R_1^2} - \frac{1}{R_2^2}\right) \quad \ldots\ldots (4)$$

(*Concluded on Page 182*)



Roller Chains Provide Unique Propulsion Drive

By N. C. Bremer
Chief Engineer
Morse Chain Co.
Ithaca, N. Y.

SUCCESSFULLY used on diesel-powered equipment for many years, transmission chain offers considerable latitude in design because of its accommodation to varying center distances and its ability to carry heavy loads under adverse conditions of protection, lubrication and alignment. Performance of several chain drives on small dual-engine commercial vessels encouraged the Navy's Bureau of Ships in 1942 to make a thorough study of the possibilities of the chain drive as a substitute for electric and helical-gear drives which, at the time, were on the critical list as regards availability. The study resulted in a decision to equip the YT 65-ft tugs and a number of the YT 110-ft harbor tugs with dual engines and chain drives connecting them to the large slow-turning propellers. Success of the drives in these boats and in the Army Transportation Corps' small FP cargo vessels has created considerable interest in this new marine transmission.

Primary function of chain drives in diesel-propelled vessels is to

SHORTAGES of helical gears and electric drives during the war led to the adoption of roller chains for propulsion drives of certain small diesel-powered ships. Experience with these drives and the advantages and limitations of transmission chains for such service are discussed in this article, which is based on a paper presented at the recent meeting of the Oil and Gas Power division of the A.S.M.E. in Milwaukee

Fig. 1—Left—Heavy-duty chain drive for transferring the power from two engines to a single shaft

transfer power from two or more engines to a single propeller shaft, *Fig.* 1. While chain drives are efficient speed reducers they will not commonly be adapted to the function of speed and torque change in a single-engine single-screw vessel, as in this case the gear drive would be indicated because of its inherent compactness.

When used with medium-speed engines the speed reduction accomplished by the chain transmission in itself is usually sufficient to allow the use of large propellers. Reversal of the propeller can be through the use of separate reverse gears on each engine, *Fig.* 2, or in the output train, *Fig.* 3, or with specially designed reversing chain drives, or by the use of direct-reversing engines, *Fig.* 4.

Preference is for drives designed for the lower revolutions-per-minute brackets, *Figs.* 2 and 4, which incorporate the heavy-duty series of manufacturers' standard steel-finished roller chain. This series covers ¾ to 2½-in. pitch chains usually made up to quadruple width as standard and 5 to 8 strands wide as special. Capacities up to nearly 2000 hp per engine may be handled by a single transmission, provided that input speed is sufficiently low.

Multiple-strand chains for marine drives should be constructed with the center plates shaved or bored to allow a heavy press fit on the pins. While this type of roller-chain design requires special tools for assembly or disconnection, it provides a "preloaded" condition of the chain side bars which is most conducive to maximum load-carrying capacity.

In a theoretical sense the maximum power-carrying capacity of a chain drive is determined by the point at which excessive sprocket width causes dangerous bending deflections to occur in the shaft or sprocket because of chain pull. In this discussion, however, the conservative upper limit of horsepower capacity will be considered as that of two 6-strand chains sharing a common load on the same sprocket.

Speed Ranges: It must be emphasized that chain drives, unlike gears, have sharply defined upper limits of rotative speed for any given pitch. This limit is usually established by the ability of the chain roller to withstand the impact forces created by the chordal action of the chain as it engages the sprocket. Again, speed ratings are

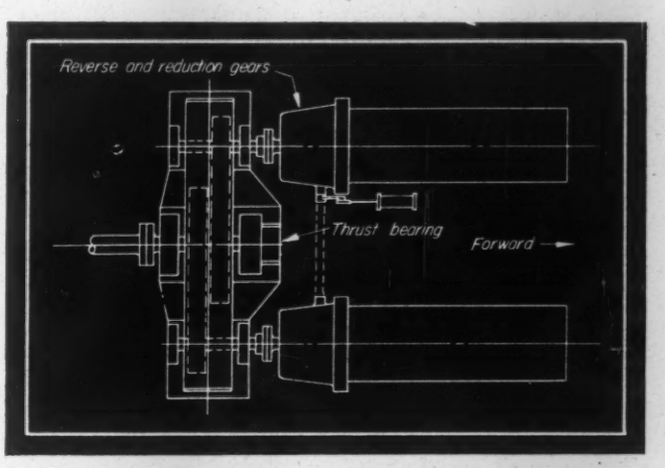


Fig. 2—Above—Twin-engine drive with reduction and reverse gears on each engine. Chain drive is low speed

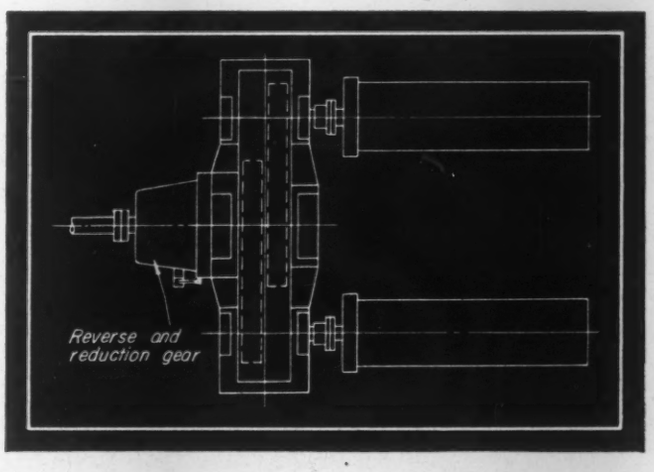


Fig. 3—Above—Drive having the reduction and reverse gears on the output shaft. Chain drive operates at higher speed than the one shown in Fig. 2

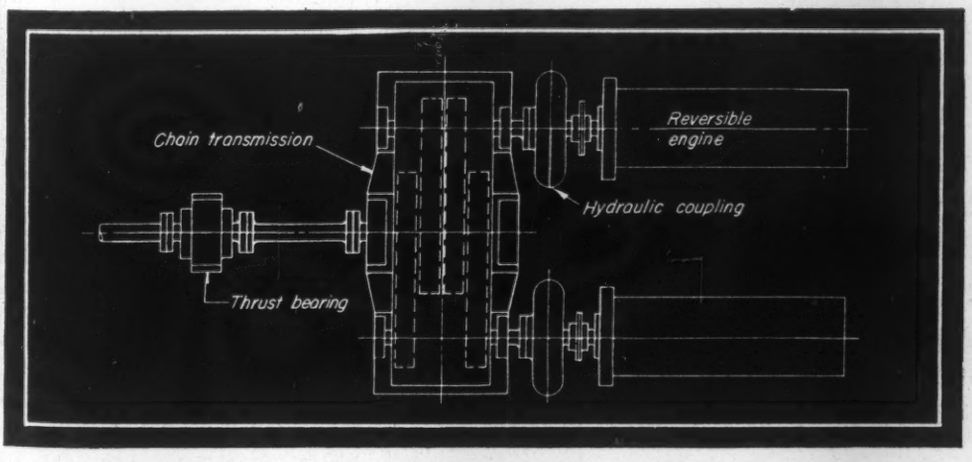


Fig. 4—Left—Use of reversible engine eliminates need for gears, entire speed reduction being accomplished by chain

conservative, assuming that all elements of the chain will have unlimited endurance capacity under continuous service conditions at full load.

While the linear speed of the chain is not a critical factor, good practice seems to dictate a range of from 2000 to 3500 fpm, the smaller pitches of chain being better adapted to the higher speeds.

CHORDAL ACTION: *Fig.* 5 shows the effect of the sprocket polygon causing chordal rise and fall of the chain. It will be noted that small numbers of teeth in the sprockets create decided variations in the velocity of the chain. The practical effect of this is to create undue noise and to set up stresses in the chain which reduce materially its endurance capacity. As the number of teeth in the sprocket increases, this velocity change drops off markedly and the quietness, smoothness, and load-carrying ability of the drive increases.

Minimum desirable number of teeth for the small sprocket of marine chain drives, regardless of pitch, is in the range between 26 and 35. The smallest sprocket in the marine drives discussed in this article contains 30 teeth. Use of this relatively generous number of teeth probably contributes more than any other single factor to the success of the drives, including quietness at all speeds.

RATIO LIMITATIONS: Ratios as high as 10 to 1 often are used in industrial drives. In these cases, however, either speeds or loads are sufficiently low to allow the use of very small sprockets. The outside limit of reduction for marine drives probably should be about 5 to 1. Even this ratio, assuming that a suitable number of pinion teeth is used, would necessitate a large driven sprocket which is seldom possible because of limited hull clearance.

When slow-turning propellers are to be used with high-speed engines, a good practice would be to choose a chain drive of about 3 to 1 ratio, using reduction gears or combination reduction-reverse gears between the chain-drive input shafts and the engines, *Fig.* 2. Heavier, more rugged chains are used with this arrangement and a better balance of power train design is possible.

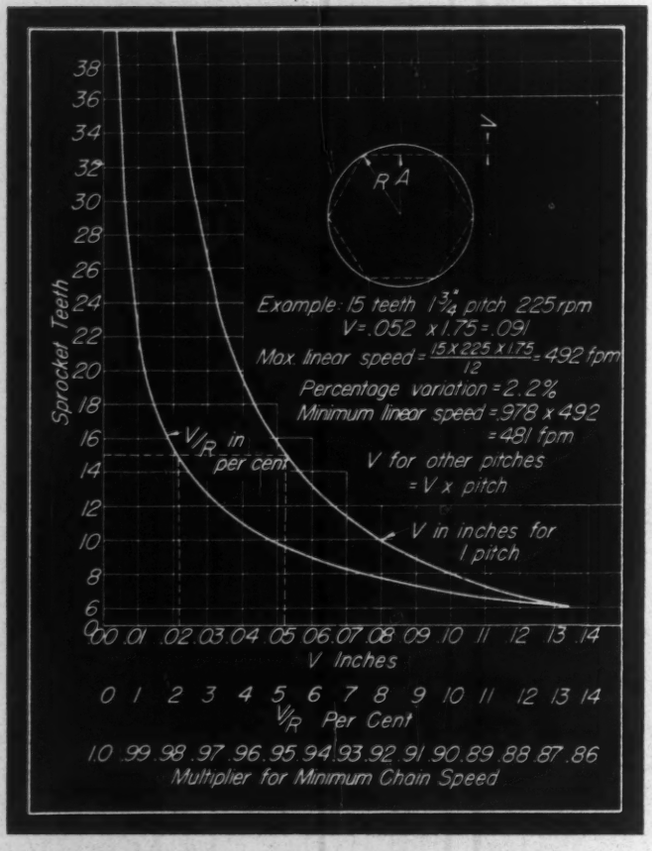


Fig. 5—Chart showing percentage variation in speed of chain during uniform rotation of sprocket

FACTOR OF SAFETY: Loads are based upon a minimum working-load to chain-strength ratio of about 1 to 30. As this high factor is necessary only because of rapidly recurring dynamic loadings created by chordal action, it is obvious that the chain has an enormous capacity for resisting momentary shocks and overloads. The well-designed marine chain transmission can withstand the effects of "rough going" as well as any other unit of the propeller drive.

ELASTIC PROPERTIES OF CHAIN: The elastic "stretch" of a roller chain due to application of the working load amounts to about 0.0006-inch per inch of its length. From this it is apparent that the natural period of the chain is quite high and little need be feared from torsional resonance with either the propeller shaft or the engines. This relative lack of "rubber" in the chains makes it necessary that where two or more chains share a common load, the chains must be carefully matched in regard to pitch to insure proper distribution of chain pull.

Lightweight Housing Design Permissible

It is also obvious that housing deflections of a nature which would create excessive parallel misalignment of the shafts must be avoided. In this respect, however, the chain drive is considerably less sensitive than a reduction gear, and chain-drive housings may be of substantially lighter construction than gear cases. Installation problems are also simpler, especially if self-aligning bearings are used.

PITCH ELONGATION—CHAIN AND SPROCKET LIFE: It is customary to set up the usual chain drive in such a manner that the "slack" caused by joint wear can be taken up. Sometimes this is accomplished by the use of adjustable idler sprockets, or more often by increasing the center distance between the sprockets with sliding motor bases or brackets.

However, it has long been established that a roller chain installed and operated under good transmission conditions develops nearly all of its "stretch" or pitch elongation during the first few hundred hours of operation and, once it has been "run-in", can be operated for thousands of hours without further elongation of any appreciable degree. Repeated tests have proved conclusively that this tendency to stretch at the start is due to the constriction at the ends of the holes in the chain bushings caused by the heavy press fit of the side bars which does not allow full utilization of the total bearing area of the chain joint.

Development of grinding and honing processes adapted to the larger sizes of chains has made it possible to assure full seating of the full length of the chain bushing bore at the very start. In addition, this also allows correction of inaccuracies of side-bar hole spacing and bushing-bore parallelism which would cause uneven loadings.

For this reason the Navy drives were designed without

(*Concluded on Page* 184)

MACHINE *Editorial* DESIGN

Research—Key to Improved Design

WITHOUT research, a modern manufacturing concern is in somewhat the position of mankind without medical aid. As Roland P. Soule puts it, industrial research performs three functions closely related to those of the medical profession: Curative, preventive and obstetrical. The first has its parallel in the curing of troubles which may arise in designs already developed; the second is akin to the improvement, both functional and in production, of current models; while the third corresponds to the bringing forth of entirely new products.

Although the word research conjures up a mental picture of Ph.D's. huddled over intricate apparatus from which they coax the innermost secrets of nature, the dictionary meaning is simply "a searching for something, especially with care or diligence". In his search for the best materials, the proper mechanisms and controls and the best production methods for his new machine, the designer is utilizing the research approach just as much as does the more highly glamorized scientist.

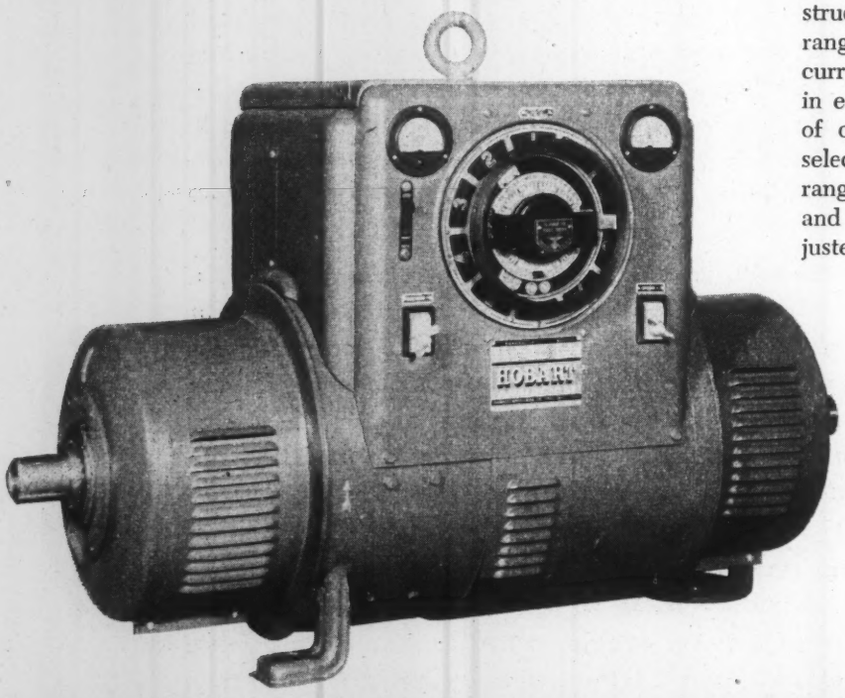
There is, however, this difference—that the scientist is much better equipped with laboratory tools which, to use a Ketteringism, are not for the purpose of solving problems but to change a mental attitude. The designer too often is thrown completely on his own resources, depending as he does on his basic knowledge and his experience to formulate in his mind the concept of a new machine or its mechanism and construction. When the designer has access to modern research equipment to confirm or refute his mental attitude, the possibilities of significant improvements in design become almost limitless.

With the intensely severe competition that will be felt as soon as present shortages ease and the laws of supply and demand once more control prices, no machinery manufacturer can afford to be without a comprehensive research program. The logical place for such activity to begin is under the auspices of the engineering department where the results will do the most immediate good, particularly in assisting the designers charged with the responsibility for the future.

DESIGNS OF THE

Synchronous Motor Welder

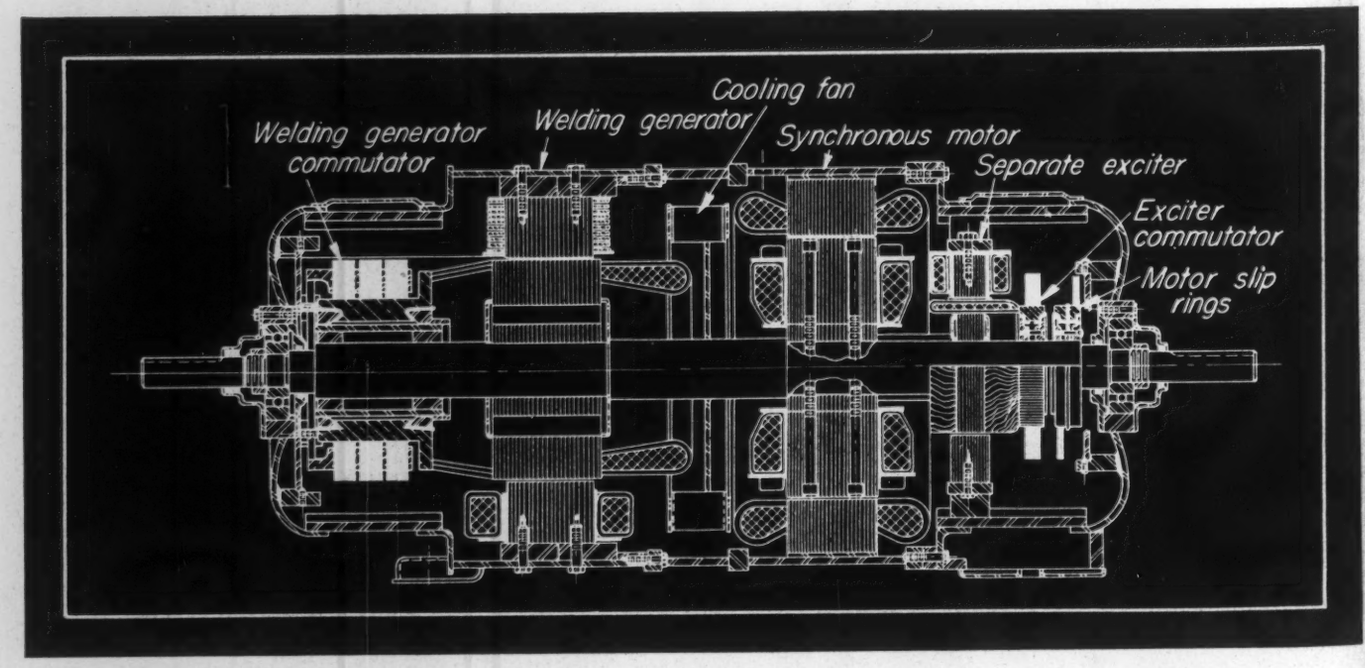
DESIGNED to draw unity power-factor current from the line, this motor-driven welder employs a revolving-field synchronous-type motor with a heavy squirrel-cage winding to make easy starting as a conventional induction motor. The separate exciter employed builds up and supplies current to the revolving fields and automatically changes operation of the motor from induction to synchronous. This excitation is supplied through two slip rings, total current varying from five to ten amperes.



Both the exciter brushes and slip-ring brushes are held in a brush-holder of unit design, simplifying construction and electrical connections. Multiple-range dual control with ten ranges of welding current and 100 steps of volt-ampere adjustment in each range make available 1000 combinations of open-circuit voltage and welding current for selecting any desired arc characteristics. Main-range switch is heavy copper molded in bakelite and controlled by the handwheel. Volt-amp adjuster is compactly built within the main switch and easily removed if remote control is desired. The polarity switch, which carries only the exciting current, is conveniently located on the front panel.

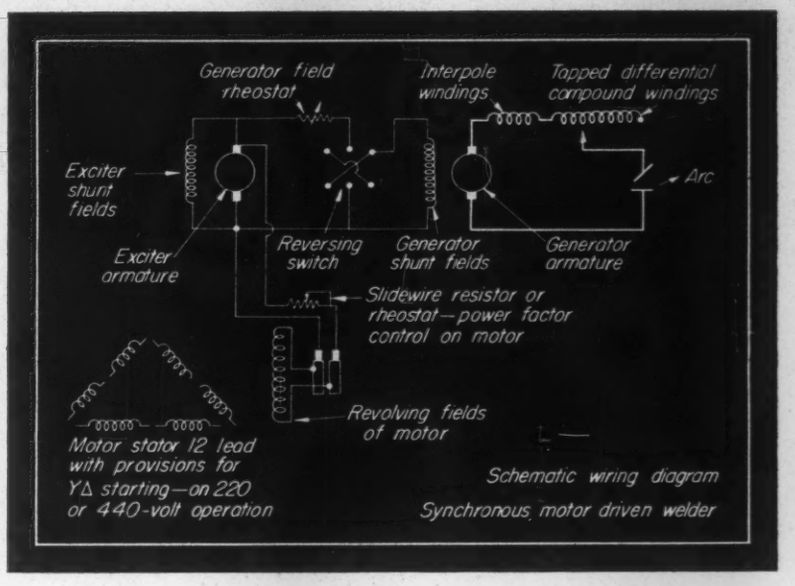
For ventilation, a special fan draws fresh air in at both ends of the motor-generator unit and expels it at the center through louver ventilators, changing the air twice as often as in end-to-end ventilating systems.

Frame of the machine is arc-welded steel, built in three sections which are bolted together for easy disassembly.



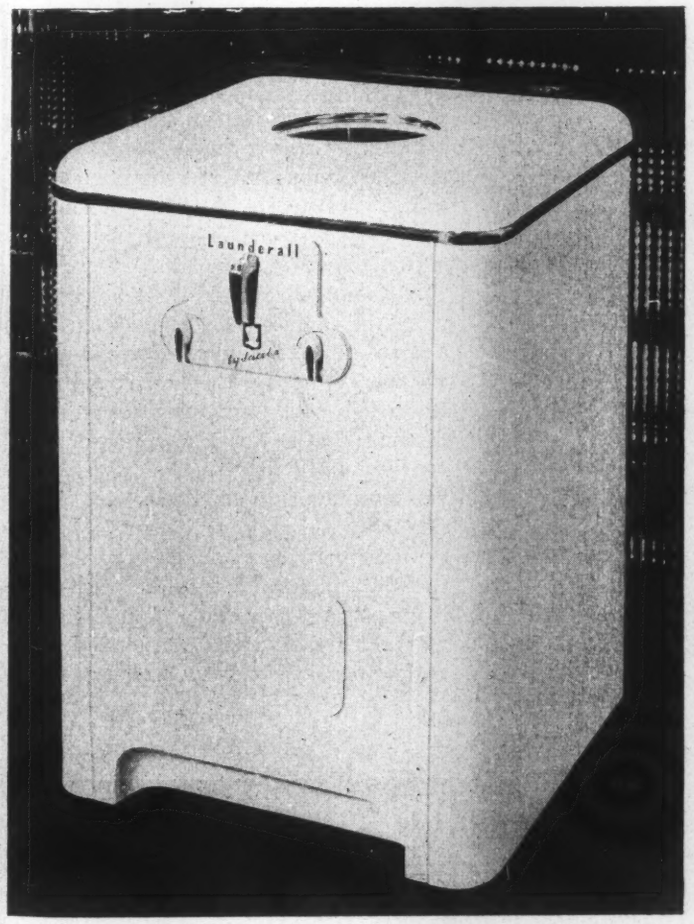
EMONTH

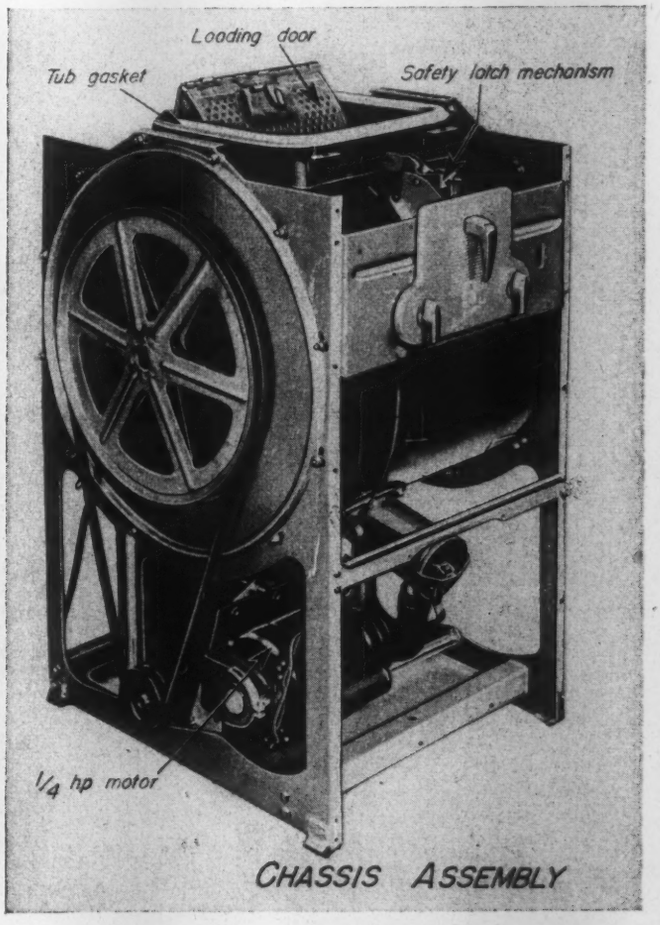
Shaft is SAE 1045 steel. The motor and generator with fan and exciter are built in a single unit and mounted between heavy-duty ball bearings. Nothing overhangs these bearings and they are readily accessible. Turret top of pressed and welded steel houses the range switch, motor starter, etc. Connectors for cables and controls are all located conveniently on the outside of the cabinet. Lifting eye makes handling convenient. Manufacturer: The Hobart Brothers Co., Troy, O.



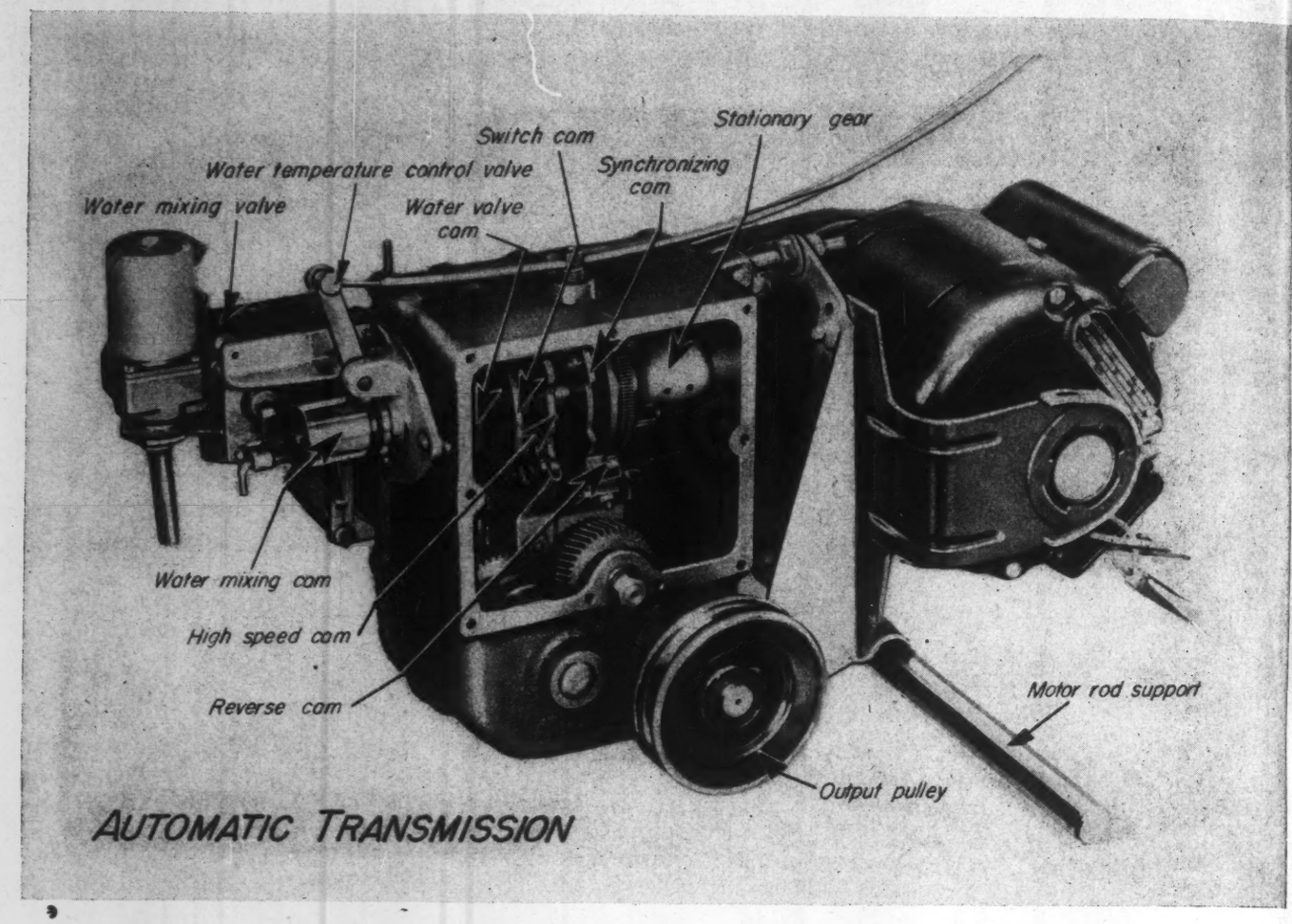
Automatic Home Laundry

CHASSIS of this automatic home laundry consists of a water compartment, or tub, and two A-frames, one at each side of the water compartment. This design permits the washing cylinder, inside the tub, to be axially supported on both ends. The tub, porcelain enameled, is a rolled sheet with a series of bolt lugs welded around its openings. It carries a drain chamber and additional screen equipment for trapping any lint or foreign matter that might clog the pump line of the machine. Tub and A-frames are bolted together and made water-tight through

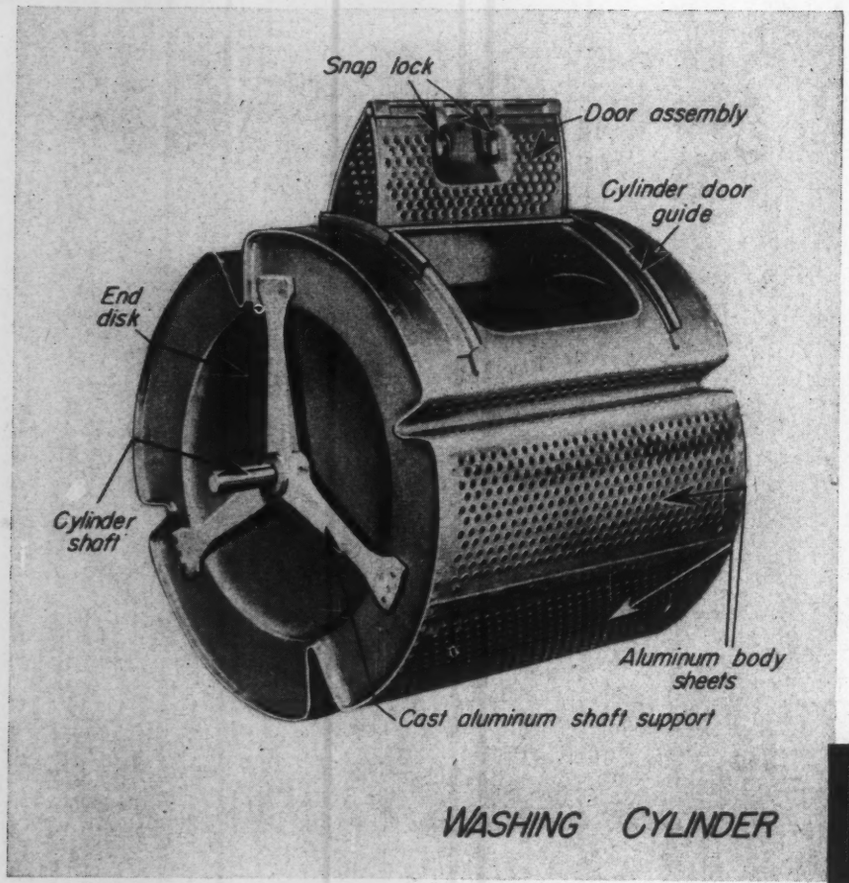




CHASSIS ASSEMBLY



AUTOMATIC TRANSMISSION



WASHING CYLINDER

use of a synthetic gasket on each end of the tub opening. Each A-frame carries a new type of rubber-sheathed bronze bearing for the cylinder shaft. These bearings, imbedded in synthetic rubber, were developed by the Navy during the war and one of their desirable features is that they require no lubrication.

A two-speed transmission is employed, with the slow-speed drive designed to operate bi-directionally. Drive is clutch-powered direct for the high-speed cycle. Reverse-direction rotation is accomplished through simple gearing, spring-clutched synchronously to give the proper number of rotations in each direction. Actuation of these spring clutches is through cam energization of clutch keys. A two-speed cam shaft initiates the wash-drain-dry and shut-off cycling. The water-mix and meter valve comprises two independently operating valves, each having an

DESIGN

individual intake from the supply—one hot, the other cold. Mixing of water to washing temperature is accomplished thermostatically.

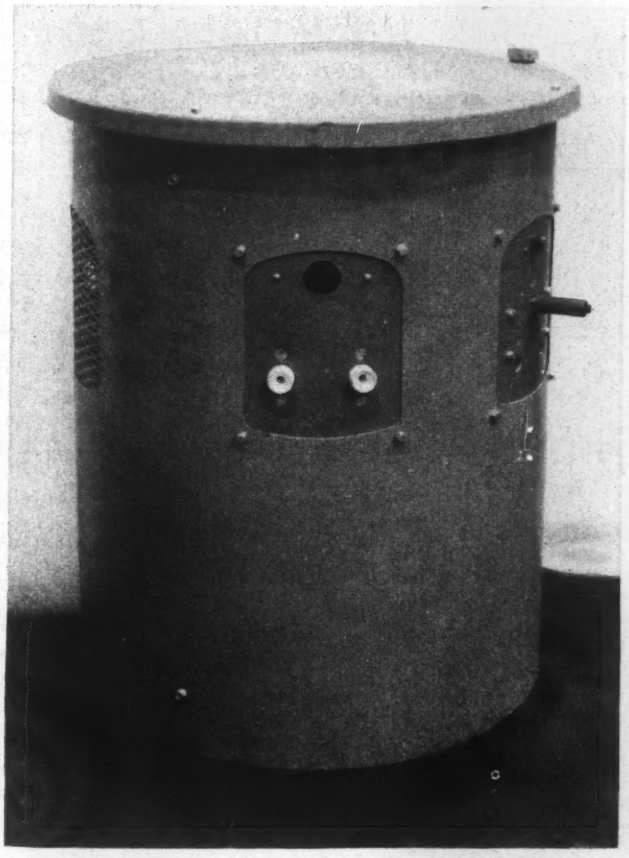
Body of the washing cylinder comprises two aluminum alloy sheets spot welded together to form a continuous cylinder with a large rolled-edge opening. Inside the cylinder are four perforated baffles designed to give turbulent water action through the wash load. A flanged disk is spot welded to each end of the cylinder. To facilitate loading and unloading of wash, a quick-opening, jack-knife door design is employed. Manufacturer: F. L. Jacobs Co., Detroit.

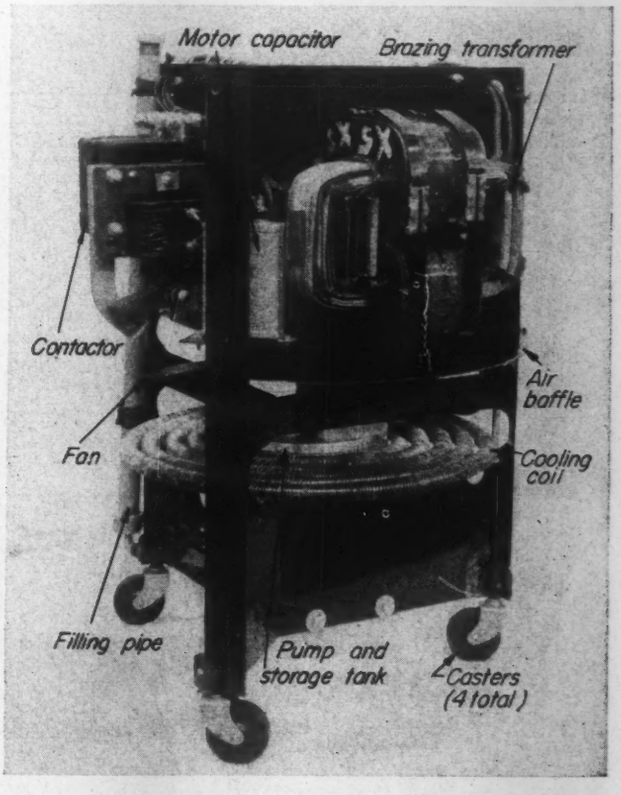
20-Kva Mobile Brazer

DESIGNED for general brazing service, this self-contained 20-kva portable brazer, enclosed in a steel housing, has a contactor for energizing or de-energizing the brazing circuit and an adjustable-heat brazing transformer. A water-cooling system is provided for the water-cooled brazing tongs supplied with the unit.

Control panel, built into the front of the unit, contains a control switch, outlets for brazing cables, receptacle for foot-switch pilot cable, and voltage tap changing plug. Electrical part of the equipment is placed above the water system so that any condensation or leaks from the water system will not drip on the electrical apparatus.

Brazing transformer is air cooled by induced draft. A fan draws the air through the transformer winding and forces it through the cooling coil. Thus the same air used for cooling the water from the tongs also cools the transformer and permits using a transformer of minimum size. Cover of the machine is solid and cooling air is drawn through the grille in the side of the case, making the unit drip proof. Control for the contactor is interlocked with the motor control so that the transformer cannot be energized unless the fan and pump are running. Nonferrous materials or stainless steel are used throughout the water system to prevent the formation of rust or scale. Entire unit is mounted on four casters. Manufacturer: Westinghouse Electric Corp., Pittsburgh.

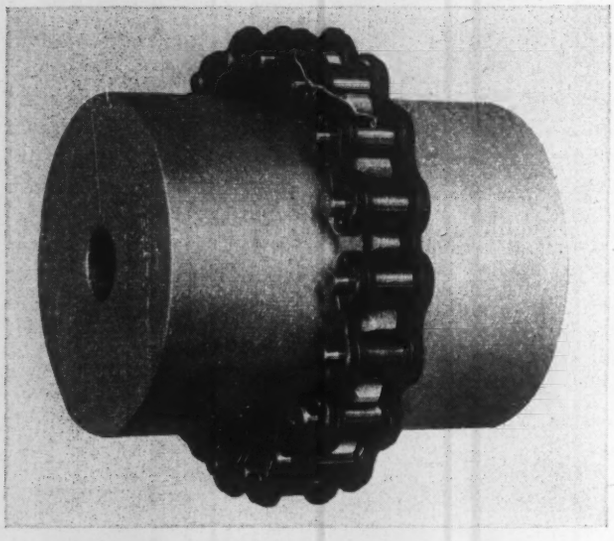




OF THE MONTH

Applications

of engineering parts, materials and processes

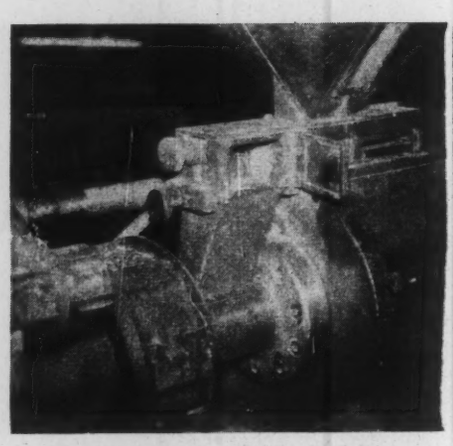


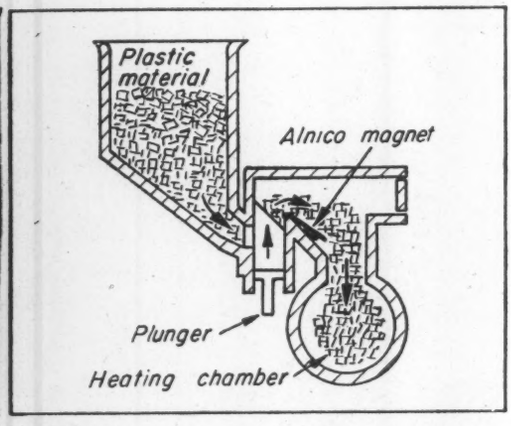
Roller Chain Makes for Coupling Flexibility

INGENIOUS use of standard roller chain as the transmission element of a flexible coupling is evidenced in the Flexoid Slow-Speed Coupling, left, developed by the Smith Power Transmission Co. Independent hubs engage the chain with alternate teeth. Functioning in this manner the chain is essentially in pure tension. By virtue of proper tooth-to-chain clearance and flexibility of the chain, angular and parallel misalignment respectively may exist without undue wear on the parts.

Eliminates Molding Problem

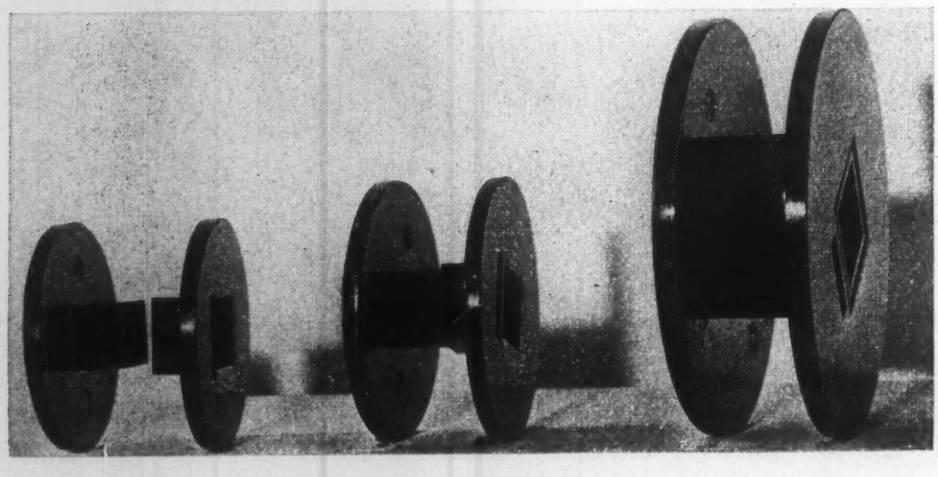
DAMAGE to dies and machinery caused by tramp iron in the raw plastic have been reduced greatly by the use of Eriez permanent magnets in the injection molder, left. Previous technique of using magnets of various types in conveyors had not been successful because of subsequent contamination. However, it was found that the plate magnet placed in the press immediately in front of the melting chamber eliminated all possible introduction of magnetic materials to the dies. Thus most mold scratches as well as their deteriorating effects are obviated.





Plastic Parts

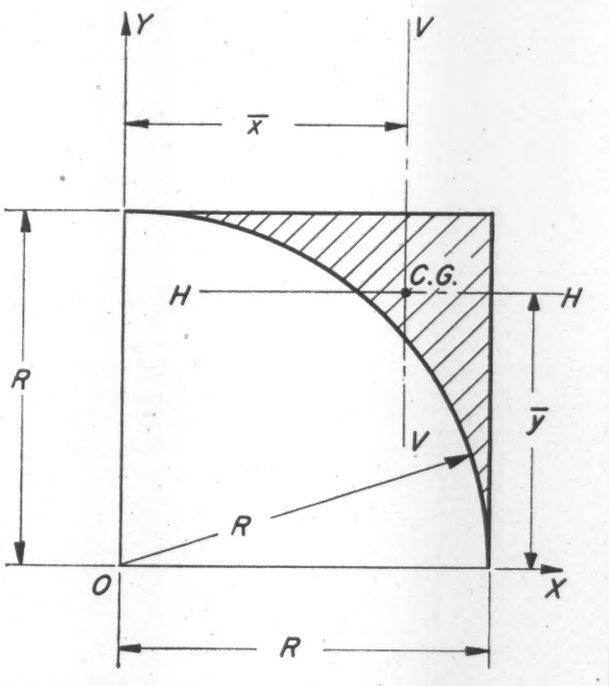
BRUSH stud insulators, left bottom, for direct-current motors, formerly made with a laminated insert, have been redesigned for molding by engineers of the General Electric Co. The modification was made possible by the use of an impact material which permitted the molding of relatively thin walls to close tolerances. This plastic material gives the necessary strength to the insulator hubs, while the ability to mold to close tolerances enabled the hubs to fit snugly one inside the other.



Section Properties of Fillets

By Victor M. Kibardin
Professional Engineer
Glen Ridge, N. J.

Part II—Fractional Fillets



Whole Fillet

Area:

$$A = 0.214601837\ R^2$$

Section properties are:

Moment about Y-axis and about X-axis:

$$M_y = M_x = 0.166666667\ R^3$$

Location of neutral axes V and H:

$$\bar{x} = \bar{y} = 0.776632061\ R$$

Moment of inertia about Y-axis and about X-axis:

$$I_y = I_x = 0.136983793\ R^4$$

Moment of inertia about V-axis and about H-axis:

$$I_v = I_h = 0.007545116\ R^4$$

Fig. 4—Exact equations for the section properties of a whole (90-degree) fillet of radius R

IN LAST month's data sheet equations, curves and tabular values were presented for the section properties of whole 90-degree and oblique fillets. The present data sheet is concerned with fractional fillets, the sectional properties of which are given in the form of exact equations. In addition, curves are presented for quick solution based on the data for whole fillets.

For whole fillets the principal section properties are given by the equations accompanying *Fig.* 4. As in the previous data sheet, numerical coefficients contain enough significant figures to satisfy any accuracy requirements.

Fractional Fillets — Case I — Fig. 5

EXACT EQUATIONS: Section properties of a fractional fillet can be determined if the length of one of the straight sides of the section is expressed as a function, k, of the radius, *Fig.* 5. The properties can also be determined if the angle, α, subtended by the arc of the fillet is known. The relations between k and α are as follows:

$$\sin \alpha = k$$
$$\cos \alpha = \sqrt{1 - k^2}$$

Accompanying *Fig.* 5 are exact equations for the section properties of the fractional fillet in terms of $\sin \alpha$ and $\cos \alpha$. If k is known, $\sin \alpha$ and $\cos \alpha$ should be calculated from the foregoing equations.

APPROXIMATE CALCULATION: Knowing the properties of the whole fillet of radius R, the section properties of a fractional fillet may be quickly determined with the aid of the multiplying factors given in *Fig.* 6. These factors represent the section properties of the fractional fillet, *Fig.* 5, expressed as a fraction of the corresponding properties of the whole fillet, *Fig.* 4, having the same radius R.

Fractional Fillets — Case II — Fig. 7

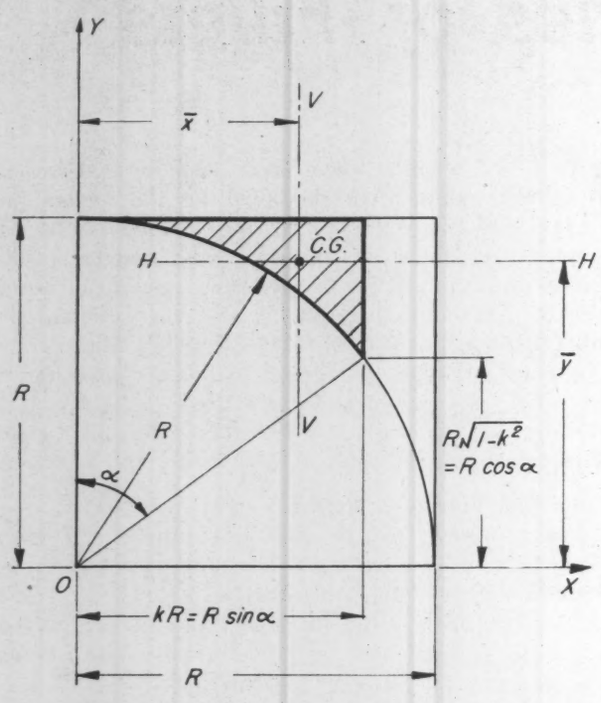
EXACT EQUATIONS: Section properties of fractional fillets complementary to Case I can be determined in a similar manner. The ratio k, measured as before, now represents the distance from the Y-axis to the variable edge, expressed as a fraction of the radius, *Fig.* 7. The angle subtended by the arc of the fillet is now the complement of the angle α, and is denoted by β, where $\beta = 90° - \alpha$. The relations between k and β are as follows:

$$\cos \beta = k$$
$$\sin \beta = \sqrt{1 - k^2}$$

Accompanying *Fig.* 7 are exact equations for the section properties of the fractional fillet in terms of $\sin \beta$ and $\cos \beta$. If k is known, $\sin \beta$ and $\cos \beta$ should be calculated from the foregoing equations.

APPROXIMATE CALCULATION: *Fig.* 8 gives multiplying factors from which the section properties of fractional fillets may be quickly determined. Each curve represents a section property of the fractional fillet, *Fig.* 7, expressed as a fraction of the corresponding property of the whole fillet, *Fig.* 4, having the same radius R.

ENGINEERING DATA SHEET



Fractional Fillet — Case I

Area:

$$A=\frac{1}{2}R^2\left[\,2\sin\alpha-\sin\alpha\cos\alpha-\alpha\,\right]$$

Section properties with respect to Y and V axes are:

$$M_y=\frac{1}{6}R^3\left[\,3\sin^2\alpha-2+2\cos^3\alpha\,\right]$$

$$\bar{x}=M_y/A$$

$$I_y=\frac{1}{24}R^4\left[\,8\sin^3\alpha-6\sin^3\alpha\cos\alpha+3\sin\alpha\cos\alpha-3\alpha\,\right]$$

$$I_v=I_y-A\,\bar{x}^2$$

Section properties with respect to X and H axes are:

$$M_x=\frac{1}{6}R^3\sin^3\alpha$$

$$\bar{y}=M_x/A$$

$$I_x=\frac{1}{24}R^4\left[\,8\sin\alpha-3\sin\alpha\cos\alpha-2\sin\alpha\cos^3\alpha-3\alpha\,\right]$$

$$I_h=I_x-A\,\bar{y}^2$$

Fig. 5—Exact equations for the section properties of a fractional fillet having dimensions shown in the sketch

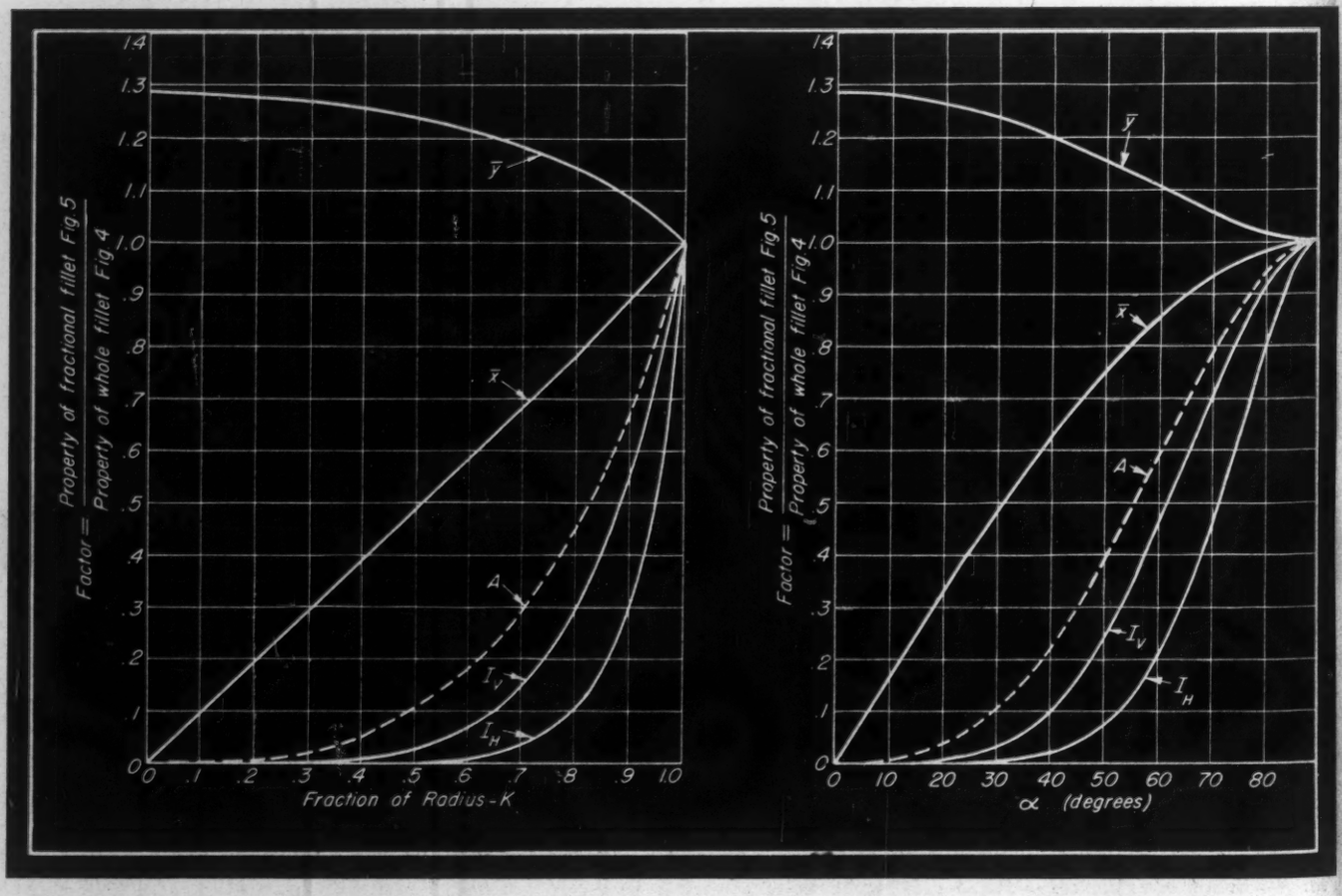
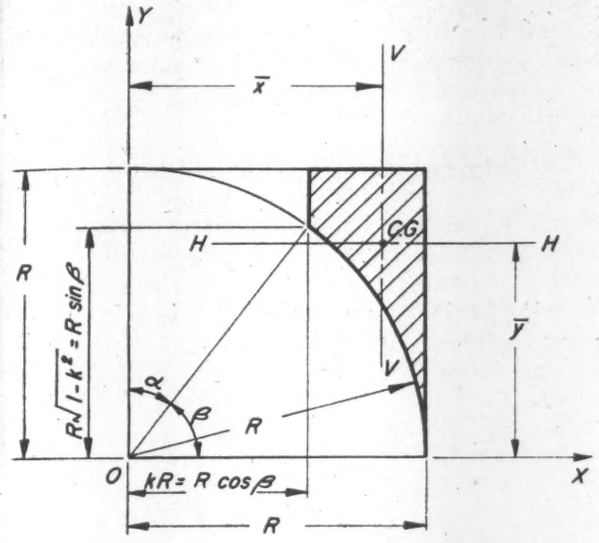


Fig. 6—Factors represent ratio of section properties for fractional fillet, Fig. 5, to those of whole fillet, Fig. 4



Fractional Fillet — Case II

Area:

$$A=\frac{1}{2}R^2\left[2-2\cos\beta+\sin\beta\cos\beta-\beta\right]$$

Section properties with respect to Y and V axes are:

$$M_y=\frac{1}{6}R^3\left[3-3\cos^2\beta-2\sin^3\beta\right]$$

$$\bar{x}=M_y/A$$

$$I_y=\frac{1}{24}R^4\left[8-8\cos^3\beta+6\cos^3\beta\sin\beta-3\sin\beta\cos\beta-3\beta\right]$$

$$I_v=I_y-A\bar{x}^2$$

Section properties with respect to X and H axes are:

$$M_x=\frac{1}{6}R^3\left[1-\cos^3\beta\right]$$

$$\bar{y}=M_x/A$$

$$I_x=\frac{1}{24}R^4\left[8-8\cos\beta+3\sin\beta\cos\beta+2\cos\beta\sin^3\beta-3\beta\right]$$

$$I_h=I-A\bar{y}^2$$

Fig. 7—Exact equations for the section properties of a fractional fillet having dimensions shown in the sketch

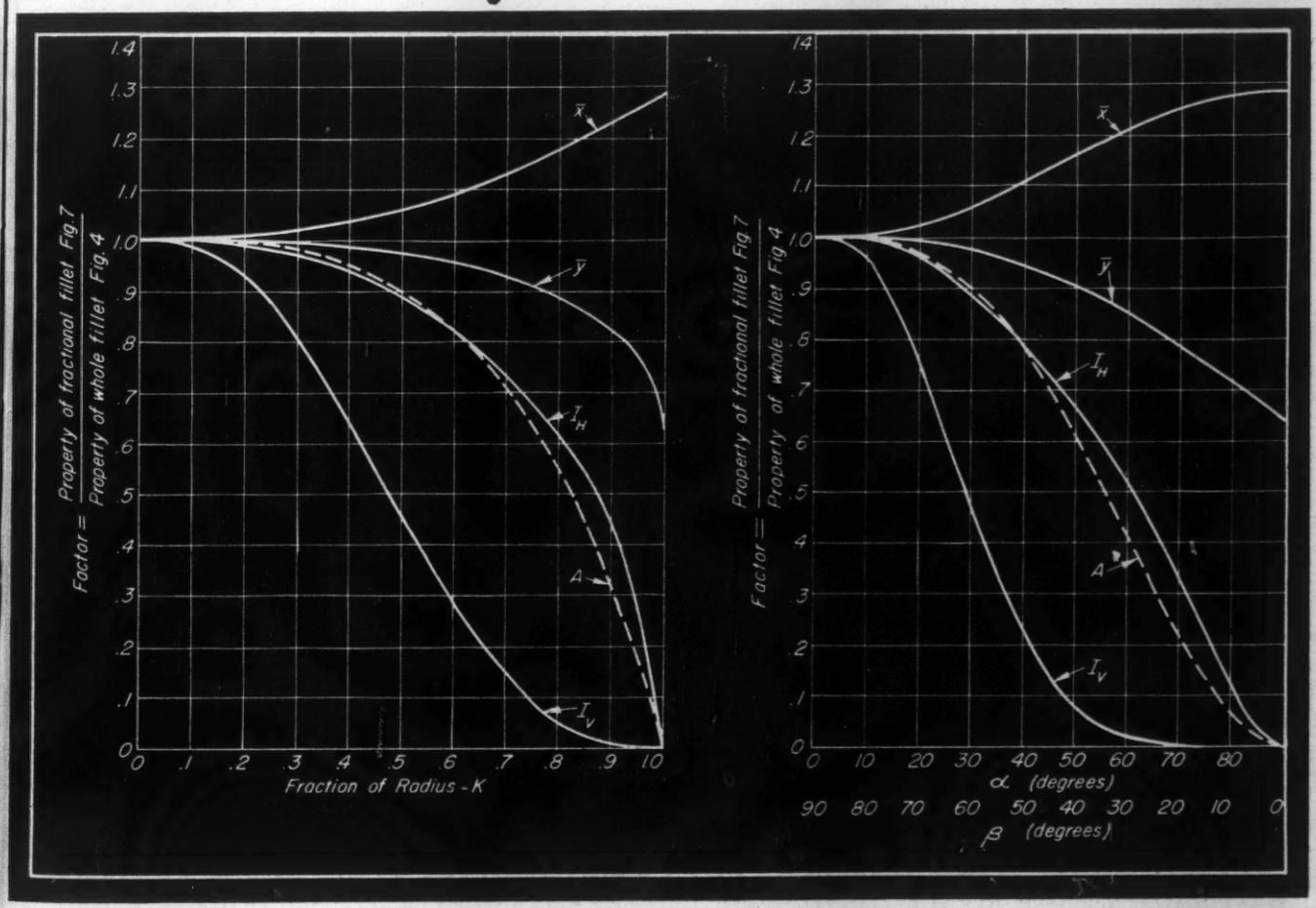
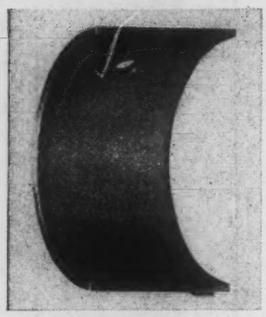


Fig. 8—Factors represent ratio of section properties for fractional fillet, Fig. 7, to those of whole fillet, Fig. 4

new parts and materials

New Grid Bearing

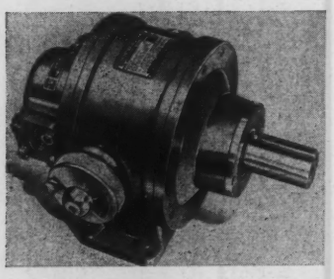


A GRID TYPE bearing combining good surface properties and good embedability with high strength and fatigue resistance has been placed in production by P. R. Mallory & Co. Inc., Indianapolis, Ind. The bearing is made by bonding a copper or silver surface to a steel backing, and knurling this surface. The knurls are then filled with lead or lead-alloy and the excess machined off, leaving a surface of base metal covered with pockets of lead-alloy. The bearing is particularly recommended for extremely high bearing loads and operating conditions which may offer extreme cases of abrasion.

Protective Coating

SERIES OF aluminum finishes covering a wide range of service requirements has been announced by the Watson-Standard Company, Pittsburgh 12, Pa. Known as Alumcote the coatings are available in nine types meeting specific requirements. They are said to make possible smoother, more brilliant and durable part-finishes, and are recommended for reflectors, containers, and heaters.

Hydraulic Motor



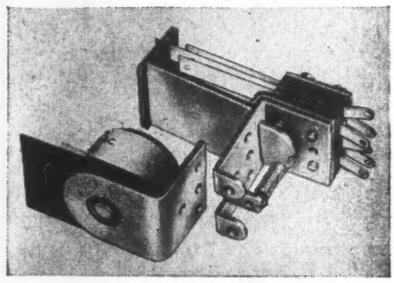
NEW TYPE of industrial hydraulic motor, combining high starting torque with "turbine-smooth" power, has been announced by Superdraulic Corp., Dearborn, Mich. Known as the Triport motor the unit parallels the line of high-pressure hydraulic pumps manufactured by the same source thus making available a balanced combination of pumps and motors. Producing 47 hp, the motor is only 10½ inches in diameter and is light enough to be handled by one man. Featuring infinitely variable speed control, forward and reverse, the motor is available in two types. One is rated at 26½ gpm, 3,500 psi input at 1,200 rpm; the other at 19 gpm, 5,000 psi at 1,200 rpm.

Spring-Driven Stepping Switch

ELECTRO-MAGNETICALLY actuated, a spring driven stepping switch has been announced by the C. P. Clare & Co., 4719 West Sunnyside Ave., Chicago 30. Operating by a pawl and ratchet mechanism, the switch will select channels or circuits from groups of 20 or 40. It may also be used for counting, totalizing or automatic control of operations in a predetermined manner.

Interchangeable-Part Relays

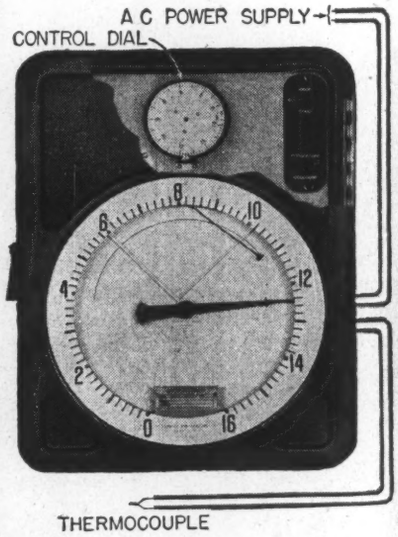
RECENTLY announced, a new type relay is being manufactured by the Guardian Electric Mfg. Co., 1601 West Walnut St., Chicago 12. The relay consists of two basic parts, a coil assembly and a contact assembly. These are fastened together with screws and lockwashers, and part



interchange is therefore relatively simple. An important feature of the relay is that the complete assortment, known as Series-200, permits of many separate relay combinations. This is possible by making different connections on the switch blade terminals and putting together any of nine standard ac or dc coils with either of two switch assemblies.

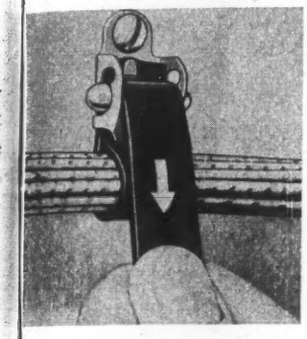
Potentiometer Pyrometer

ELECTRONIC potentiometer pyrometer, having no continuously moving or vibrating parts in its measuring circuit, is announced by Bailey Meter Co., 1050 Ivanhoe Road, Cleveland 10. The instrument, which operates on 115 volts ac may be controlled from a thermocouple or any source of dc potential varying through at least 10 millivolts for full scale range. Conventional thermocouples may be used for temperatures up to 3000 F. For temperatures above 3000 F, radiation type thermocouples



are used. Instead of a galvanometer or millivoltmeter, the Bailey Pyroton potentiometer circuit employs a dc to ac converter and an electronic detector to measure potentiometer unbalance. Temperatures are recorded by the instrument on a 12-inch diameter uniformly graduated scale. Temperature indications are shown on a 29-inch scale encircling the recording chart.

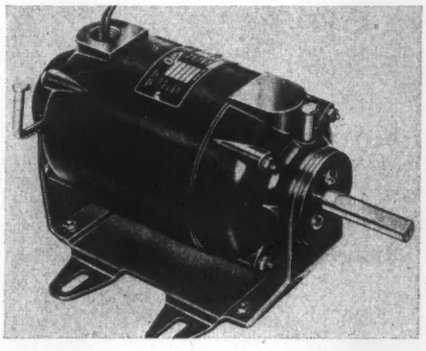
Harness Clamp



STRAP-TYPE harness clamp for aircraft wiring has been developed for greater flexibility of application. Because the clamp will handle a wide variety of cable diameters, the need is avoided for stocking wide lots of clamp sizes. In use the strap is looped around the bundles of wire and then slipped under the loose-locking roller in the bracket. The strap is then pulled up tight in the opposite direction, and the roller forced into locked position. Strap is neoprene-coated fabric to withstand the temperature range of minus 65 to plus 160 F. The unit is manufactured by Tinnerman Products Inc., 2085 Fulton Road, Cleveland 13.

Continuous-Duty Motor

MANUFACTURED by the John Oster Manufacturing Co., Terminal Bldg., 1 Main St., Racine, Wisconsin, is a new fractional-horsepower motor recommended for quiet, vibrationless, long-life operation. Known as type KS-837,



the motor delivers 1/10-hp at 1725 rpm. It is a split-phase, totally-enclosed, induction type operating on 115 volts ac. Measuring 8⅜-in. long and 4 27/64-in. in diameter, the unit is available in flange or neoprene mounting. Weight is 15 pounds.

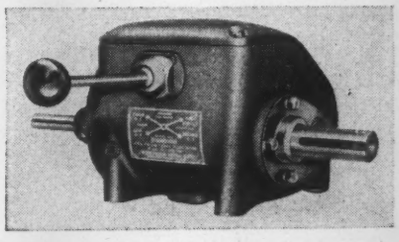
Variable Speed Transmission Drive

PROVIDED with a built-in electric drive motor, the Portman Model HT2 transmission will supply infinitely variable speeds to 1725 rpm forward and reverse. Manufactured by the Portman Machine Tool Co., 70 Portman Road, New Rochelle, N. Y., the unit is available in 2, 3 and 5 hp ratings. It is said to provide constant torque regardless of speed, and give maximum torque on starting,

thus being specially suited to high inertia loads. Measuring 13½ inches wide and 19½ inches long the drive may be obtained with standard handwheel control, or special controls to suit specification.

Four-Speed Transmission

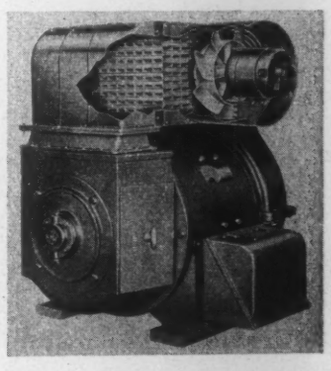
ESPECIALLY adapted to flat-plate or platform mounting, a new design of Western transmission is announced by the Western Manufacturing Co., 3400 Scotten Ave., Detroit 10. Identified as the 9000 B Series, the unit has 5-hp



capacity with input speed of 900 rpm. Four gear changes are available through one shift lever of the automotive type. The case is provided with four drilled and tapped bosses, so that the transmission can be secured to a plate or a flat portion of the machine on which it is to be mounted.

Unit-Cooled Motor

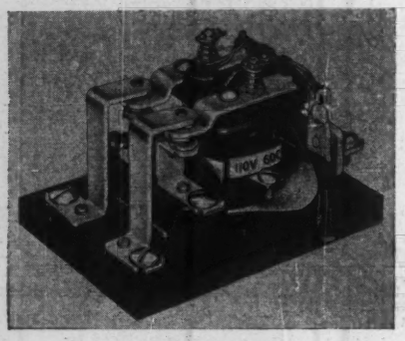
LINE OF unit-cooled, totally - enclosed, direct-current motors especially designed for use in severe atmospheric conditions has been announced by the General Electric Co., Schenectady, N. Y. Available in sizes 15 to 200 hp the motors are ventilated by a system similar to that of an automobile radiator. The unit cooler has



two sets of air passages arranged at right angles to each other. The fan on the armature circulates air through the motor and one set of passages to form a completely enclosed system. External air, maintained at a constant rate of flow by an auxiliary motor fan, travels through the other set of passages to absorb heat. A totally-enclosed construction is thus achieved without external duct work, piping, air filters or pressurized air supply.

Power-Type Relay

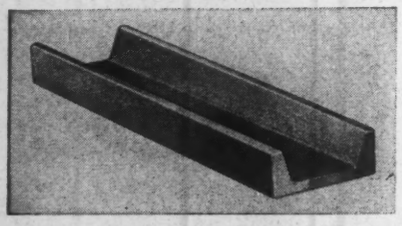
DOUBLE-POLE double-throw power relay known as Type PR-8 has been announced by Potter & Brumfield Sales Co., 549 West Washington Blvd., Chicago 6. Especially designed for such power circuits as motor starting and heating loads up to 15 amps, the relays demand only



4 volt-amperes or approximately 3 watts for operation. Metal parts are heavily plated for corrosion resistance and heavy insulation is used throughout. The relay is designed with full-floating contact pressure to reduce chatter, proper shading to prevent hum, and coils do not heat on continuous duty.

Stainless Channels

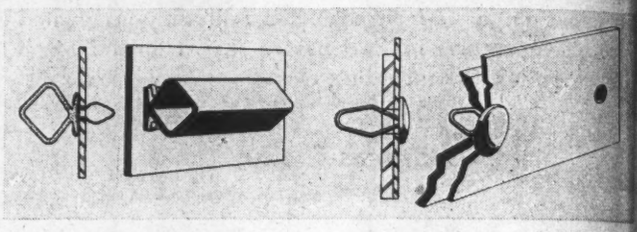
HOT-ROLLED stainless-steel channels in a variety of shapes have been announced by the Joslyn Mfg. & Supply Co., Fort Wayne, Ind. The channels which are annealed



and pickled in A.I.S.I. Types 302, 304 and 316, are available from ¾ x ⅜ x ⅛-inch to 2½ x ⅝ x ¼-inch in any length desired.

Snap Clip

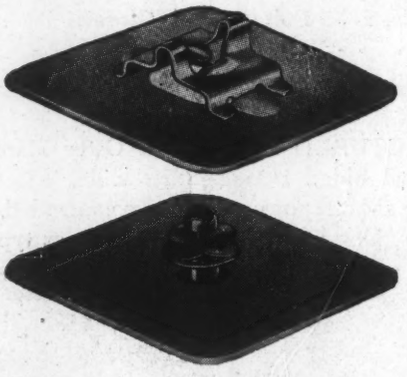
SIMPLE time and labor saving fastener which engages instantly without the use of tools is announced by the Cuyahoga Spring Co., 10256 Berea Road, Cleveland. The clips are made from either round or flat wire and are adaptable for joining parts of wood, metal, plastic, fabric, glass, etc. Installation is one-man operation, a slight pressure of the bare hands fitting the snap-clip snugly into



place. A tight permanent grip is said to be maintained by the patented "expanding feet." The clips are recommended for use in attaching trim, molding, panels, tubing, conduits, etc.

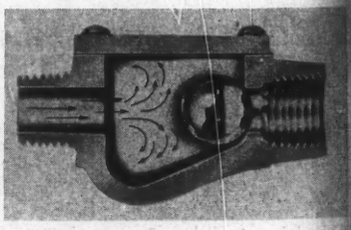
Spring-Locking Fastener

KNOWN AS Q-Two, a simplified, light duty quick-operating fastener has been announced by Shakeproof, Inc., 2500 N. Keeler St., Chicago. Designed for use on removable and hinged panels wherever accessibility for inspection, oiling, adjusting, cleaning or servicing is required, the fastener is said to offer low first cost and installation costs. The unit consists of three parts; a stud, a spring



washer which holds the stud on the outer sheet, and a receptacle which is easily snapped on to dimpled inner sheets or frames. Features include allowance for variations in sheet alignment up to ⅛-in., compensation for variations in sheet thicknesses, elimination of rivets and screws, and positive locking with one-quarter turn.

Safety Check Valve



ELIMINATING the danger of wildly whipping compressed-air hoses, a new Numatic safety check shuts off air immediately in the event of excess rate of flow. The valve is comprised of two main parts, the housing and a steel ball. Resting in a sump under normal pressures, the ball is forced up the slope to a seat upon excess air flow. When pressure returns to near normal, the ball drops away

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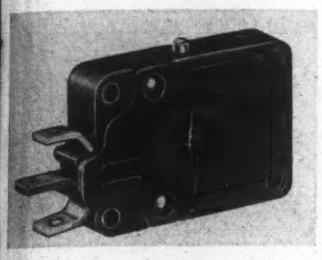
from the seat and returns to sump. Manufactured by the Hackett Manufacturing Co., the valve is distributed through the E. D. Bullard Co., 275 Eighth Street, San Francisco.

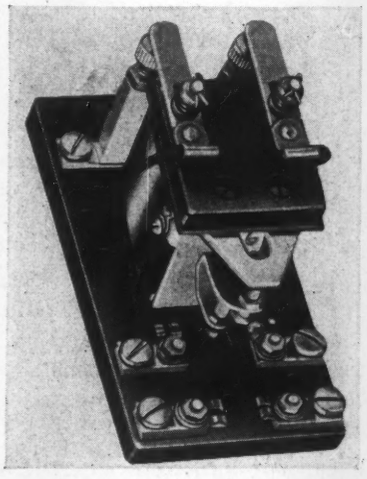
Miniature Switch



MODEL "M" Rolling Spring snap-action electric switch has been announced by Acro Electric Co., 1311 Superior Ave., Cleveland 14. It is made with single-pole, and is available in single or double-throw contact and is rated at 10 amp, 125 volt ac. The unit is made for single or multiple assemblies — four switches may be mounted in a space less than 1⅛ inches. To facilitate connections, larger barriers have been used than heretofore, and heavier solder terminals.

Heavy-Duty Relays

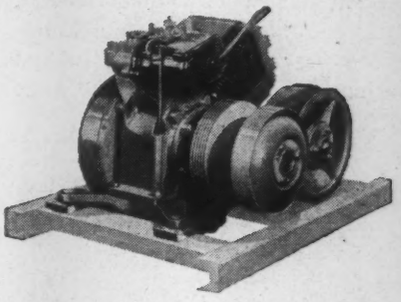
RELAYS designed for heavy duty industrial and electronic applications have been announced by the Ward Leonard Electric Co., Mt. Vernon, N. Y. Useful in connection with the control of single-phase motors, light con-



tactor duty, and automatic control purposes, the relays are available with from one to four poles, normally-open or normally-closed, single or double pole. Operating voltages are 6 to 230 volts dc, and 6 to 440 volts ac. Known as the Bulletin 130 Relays, they are manufactured with mechanical interlocks for reversing service, and mechanical latches for manual or electrical service.

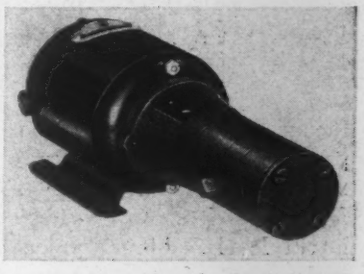
Variable Speed Power Supply

KNOWN AS the Power Package a new unit combining a 6.5-hp internal-combustion engine with an automatic transmission has been announced by Salsbury Motors Inc., 4464 District Blvd., Los Angeles. The 6.5-hp, 3200-rpm engine is coupled to the output shaft by a variable-diameter drive pulley, a variable-diameter driven pulley and a V-belt. These transmission units form a co-ordinated assembly which gives automatic infinitely-variable ratios over 4 to 1 range. Built into the drive pulley of the unit is a Salsbury clutch, an automatic, centrifugally-operated, opposed-shoe type clutch to provide smooth engagement or disengagement according to the speed of the engine. The unit is said to possess high starting torque and un-



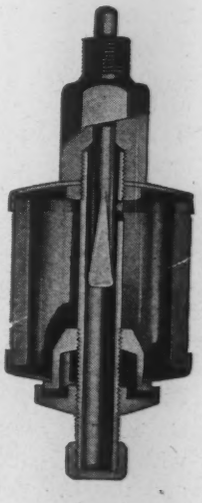
usual advantages where momentary over or under loading occurs. It is adaptable to a variety of needs in stationary, portable, or vehicular fields.

Vane-Type Pump



ANNOUNCED by the Western Engineering Co., New Haven, Conn. a new vane type pump has been designed for nonlubricating liquids such as water. The pump known as Type VW-1 has two fluid-lubricated composition bearings and vanes made of the same material as the bearing itself.

Rod Bottle-Oiler



ANNOUNCED by the Oil-Rite Corporation, South 13th St., Milwaukee 7, is a vibrating-rod bottle oiler for use on sleeve bearings or spindles. Said to be particularly suitable for dusty surroundings, and where light or medium-bodied oils are satisfactory, the oiler may be refilled without removing it from its installation. The reservoir is made of heavy-wall unbreakable Lucite, with metal parts of brass. Seven capacities from ½-oz. to 32-oz. are available, with pipe sizes ranging from ⅛ to ½-inch. Feed rod rests direct on shaft.

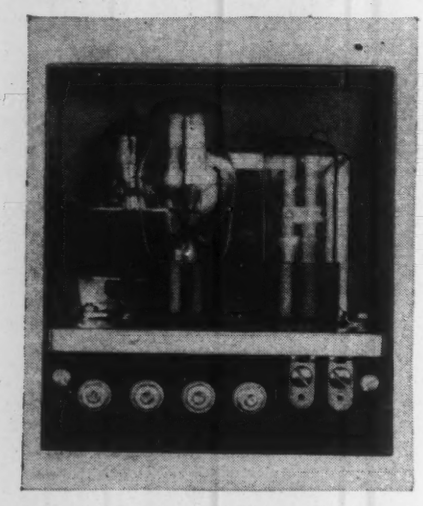
Low-Temperature Brazing Alloys

NEW silver brazing alloys featuring low silver content have been announced by Handy & Harman, 82 Fulton St., New York 7. Named Easy-Flo 45 and Easy-Flo 35, they

are 45 and 35 per cent silver, respectively. The former has a melting range from 1120 F to 1145 F while the latter has a melting range 1115 F to 1295 F. Both alloys are said to be free flowing and to produce strong ductile joints between ferrous, nonferrous and dissimilar metals.

Explosion Proof Electronic Switch

A NEW electronic switch, Model 8336, has been announced by the United Cinephone Corp., Torrington, Conn. Embodying a relay having hermetically sealed contacts, the unit has been designed to meet the demand



for an electronic switch to be used in the presence of explosive gases, and retaining the features of the standard 30 amp Cinephone switch. The switches are used primarily for pressureless limit switching, and floatless control of liquid levels. Maintainance other than the infrequent replacement of the Thyratron tube is not required.

Diaphragm Cloth

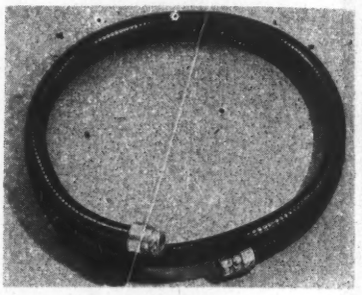
PRODUCED by the Irvington Varnish and Insulator Co., 6 Argyle Terrace, Irvington 11, N. J., a new diaphragm cloth is said to have outstanding flexibility through wide temperature ranges. Developed for use in the automotive field, the cloth is being recommended for applications where strength, flexibility, operating temperatures, and resistance to certain solvents are important considerations. Known as Style DL-011, the cloth is available in 100 to 500 yard rolls 37 inches wide.

Solenoid Type Starter

A NEW TYPE starter has been announced by the Master Electric Company, Dayton, Ohio. These starters are of the solenoid type with vertical straight-line motion, and use bridging type contacts. The unique single button feature together with an "on-off" indicating flag is said to be a marked improvement in starter design, for it simplifies the starting and stopping operations by eliminating

one push-button. One single-button remote-control station can be used with a starter of any voltage from 110 to 550 volts or frequency from 25 to 60 cycles. For applications where multiple remote control of a single starter is required the conventional two-button station is available. Starter enclosures are of heavy sheet metal, finished externally in machine tool gray and internally with aluminum lacquer. Covers are held in place with quarter turn Shakeproof fasteners.

Synthetic Covered Shielded Conduit



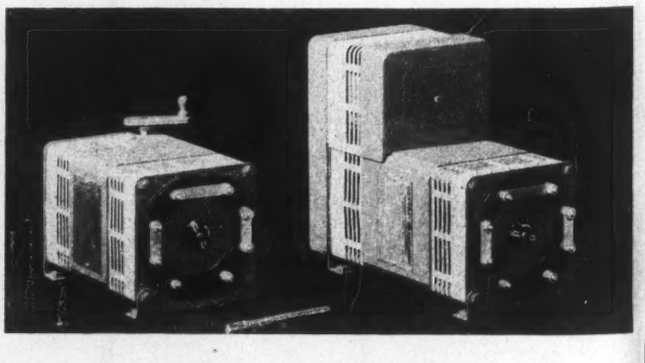
FLEXIBLE and durable, a new flexible metal hose has been added to the line manufactured by the Chicago Metal Hose Corp., Maywood, Ill. Designated as Rex machine tool conduit RT-25, the new conductor features a galvanized steel flexible metal hose liner and synthetic cover. It is liquid tight externally, usable with both standard and watertight conduit boxes, and is furnished with fittings.

Pushbutton Switch

SINGLE-POLE momentary-contact pushbutton switch for flush mounting has been announced by Ark-Les Switch Corp., 51 Water Street, Watertown 72, Mass. The switch, which is available in either the normally-open or normally-closed version, consists of a molded-plastic body and nickel-plated cap with ⅜-inch threaded stem for mounting. Identified as Part No. 7716 in normally-closed and Part No. 7716B in normally-open, the switch is rated at one amp 125 volts ac. Unit is ¾-inch in diameter and measures 1-9/32-inch overall with button protruding ¼-inch.

Induction Voltage Regulator

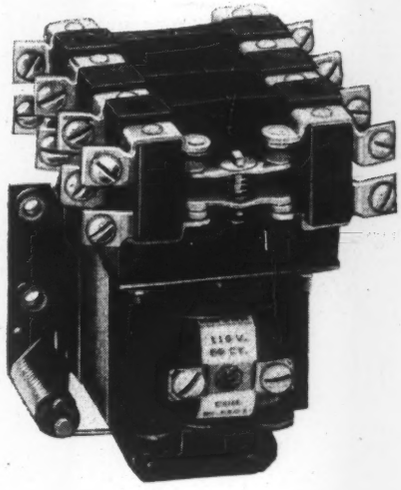
DRY-TYPE induction voltage regulators rated at 300 and 600 va have been announced by the Transformer Division of the General Electric Co., Schenectady 5, N. Y.



Developed for use in communication equipment where accurate, stepless voltage regulation is needed the new Inductrols are available in two designs: Hand or motor operated for providing voltage from a relatively constant supply voltage, and automatic for maintaining closely regulated output voltage from a varying supply voltage. Operating at autotransformer efficiency and having low radio interference the units are designed for continuous operation with a temperature rise not to exceed 55 C.

Solenoid Relay

BULLETIN 700 Universal solenoid relays have been announced by the Allen-Bradley Co., 1311 S. First St., Milwaukee 4. A solenoid type of relay having one moving part, a simple shift of connections makes possible a cir-



cuit change from normally-open to normally-closed operation, or vice versa. The relays have a maximum current rating of 10 amperes noninductive, and can be obtained for the conventional frequency ranges. Voltage ranges are from 6 to 600 volts. The relay is recommended for switching circuits of like polarity and cannot be used as a reversing switch.

Remote Switch Control Unit

REMOTE CONTROL unit combining an electrical switch and flexible control to allow for remote switch operation has been announced by Arens Controls Inc., 2253 S. Halsted St., Chicago 8. The unit is adaptable to all types of small machine tools, thereby providing lowered maintenance costs and greater operating convenience. The control has a flexible casing and may be mounted at the point most convenient for the machine operator. Available with flexible steel casings or flexible waterproof casings, the unit includes a 10 amp switch and aluminum box.

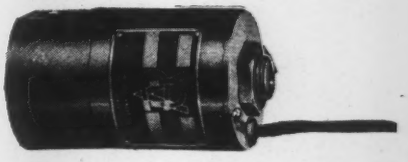
New Conveyor Idler

BELT CONVEYOR IDLER known as Rex roller-bearing style No. 55RC rubber covered spiral return idler has been announced by the Chain Belt Co., 1600 W. Bruce Street, Milwaukee. This idler is built of a flat-bar steel helical spiral, covered with rubber. The spiral composed of right and left hand sections to insure belt centering is machine-formed and permanently welded to the bearing tube. The new type idler is said to greatly retard ice forming because of rubber to rubber contact, abrasive

and corrosive action is reduced to a minimum by the elimination of metal in contact with the belt. It is said by the manufacturer that there is no build up on the spiral when used to convey sticky materials because of the shifting contact point which tends to work the material loose. The spiral idlers are used only immediately following the head pulley thus providing space for the belt to free itself of material adhering to it.

Three-Way Solenoid Valve

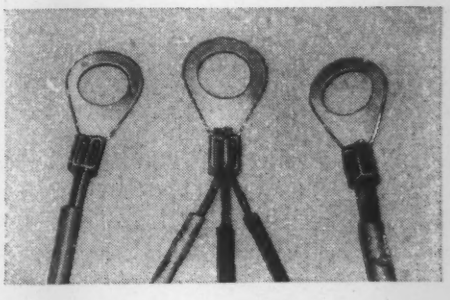
THREE-WAY, two-position solenoid valve known as V5-100 has been announced by the Allied Control Valve Co., Inc., South Norwalk, Conn. So designed as to have only



stainless steel in contact with the fluid medium, the valve is intended for 150 psi, consumes 10 watts power, weighs one pound, and is threaded to accommodate ⅛-in. pipe.

Solderless Connectors

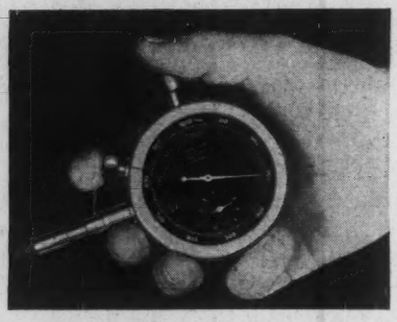
INSTALLATION tools and terminals for attaching solderless connections to solid wire, a combination of solid and stranded wire, or stranded wire have been announced by Aircraft-Marine Products, Inc., 1591 North 4th St., Harrisburg, Pa. All copper, hot electro-tinned, the ter-



minal is said to withstand vibration and torsional twist. The tensile strength of the installed connection is in excess of commercial or military requirements. Having 360-degree contact with the conductor and a short barrel, the units are available for wire sizes 22 to 10, inclusive.

engineering dept equipment

Chronometric Tachometer



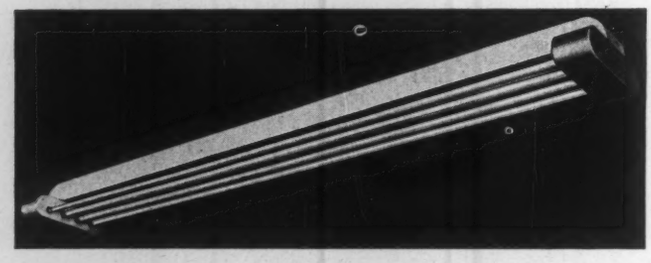
MEASURING shaft speeds from 1 to 10,000 rpm, or belt speeds from 1 to 3000 fpm a new tachometer has been announced by the Milwaukee Lock & Mfg. Co., 740 West Virginia St., Milwaukee 4, Wis. The instrument which measures in 3 second intervals is accurate with ±½ of one per cent of measured speed. Of chronometric type, the instrument is lightweight, designed to fit the hand, and nonmagnetic up to 2000 lines of flux. It is supplied in a finished case with necessary accessories including concave, convex and wheel tips.

New Sliderule

MANUFACTURED by the Frederick Post Co., Hamlin & Avondale Avenues, Chicago, is a new 10-inch plastic sliderule. Known as Plas-Ten, the sliderule features precision graduation, smooth operation, and quick, easy adjustment. It is available in a leather carrying-case.

Fluorescent Lighting Fixture

COLD-CATHODE fluorescent lighting fixtures have been announced by Colonial Lighting Co. Inc., 2901 Tonnele Avenue, North Bergen, N. J. Measuring 12½-inches wide, 8-feet long, the unit, which is available with either



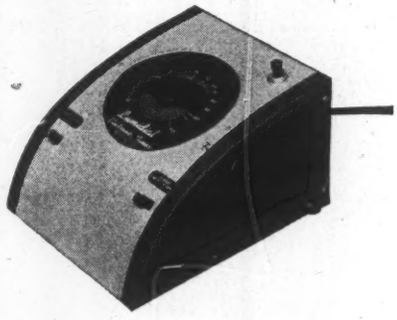
two or four lamps, is rated at 10,000 hours. The fixture is made of 20 gage steel with baked enamel finish and a wide selection of colors for end boxes. It is guaranteed for one year against mechanical and electrical defects.

New Scribing Plate

KNOWN AS the Kodak green scribing plate, a new engineering material has been announced by the Eastman Kodak Co., Rochester 4, N. Y. The new product, which consists of a transparent green-dyed gelatin coating applied to glass, permits lines to be scribed by cutting through the gelatin film. The plates may be used directly with contour comparator or as printing masters for photographically duplicating the contour comparator plate. They may also be used for making small photo-templates on metals, plastics and other materials. The plate is said to permit easier, more accurate reading by using a supplemental red light behind the plate, tolerance limits then appearing as red lines against the dark green background.

Electronic Timer

DESIGNED FOR applications requiring highly accurate time intervals, a new electronic timer has been announced by Assembly Products Inc., 1014 Bell Street, Chagrin Falls, Ohio. The timer is controlled by a continuously



variable rheostat and will give accurate time for the range 0 to 60 seconds. The unit known as the Lumidial electronic, has a luminous safelight dial, pushbutton control, and employs no clockwork or motor. It is controlled by a single thyratron tube, operates on standard 110-volt circuits, and supplies power up to 500 watts.

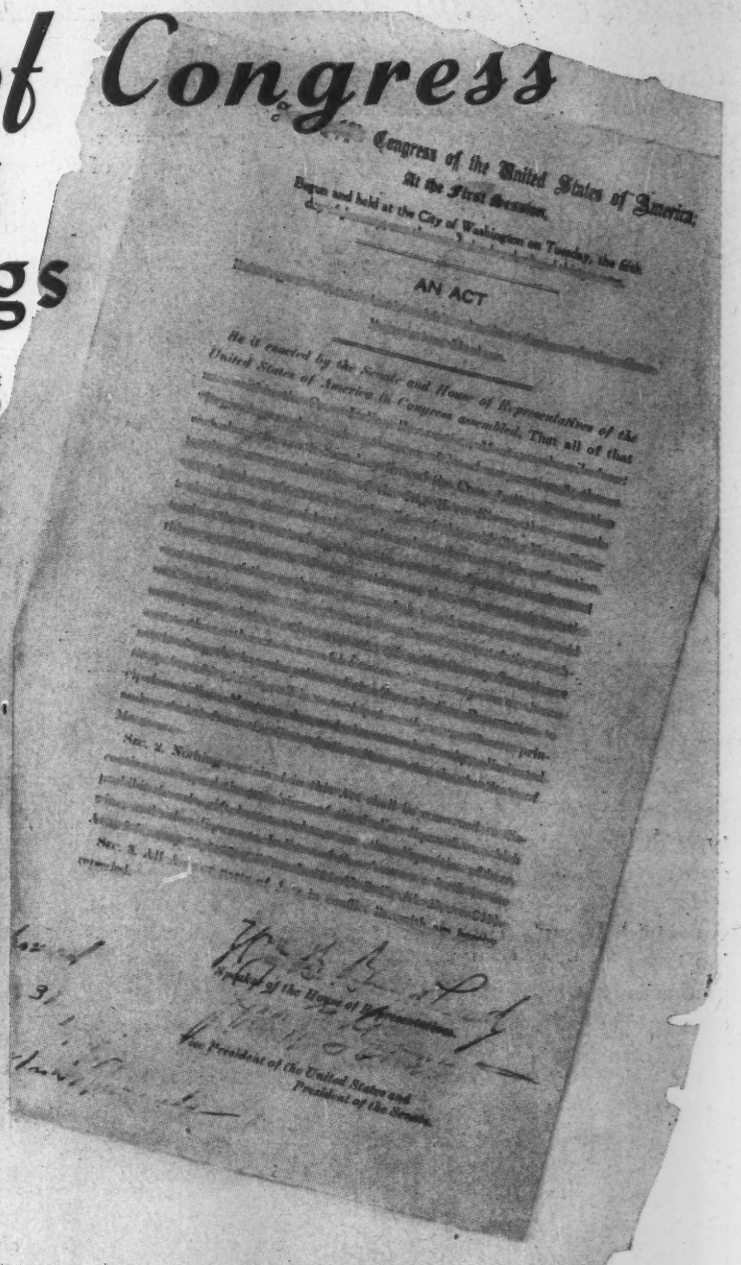
Sensitized Papers

POSITIVE prints may be made direct from drawings and tracings without the use of a negative with the new Weiline sensitized papers. Developed by a dry process, prints produced are flat and undistorted. Papers are available to give blue, red, or sepia prints on white background. The manufacturer is J. H. Weil & Co., 1315 Cherry Street, Philadelphia.

Temperature Regulator

NEW MERCURY temperature regulator with signal light, either red or white, which glows brilliantly while temperature is rising and shuts off when setting temperature is reached, has been announced by the Washington Glass Laboratory and Instrument Co., 3224 Georgia Ave., N. W., Washington 11. The signal light can be seen at a distance when glowing, and thus permits operator to perform other duties since he can determine at a glance when temperature has reached setting temperature and when

An Act of Congress concerning Sleeve Bearings


Congress of the United States of America;
At the First Session,
Begun and held at the City of Washington on Tuesday, the 6th
AN ACT
Be it enacted by the Senate and House of Representatives of the United States of America in Congress assembled, That all of that
Speaker of the House of Representatives.
Vice President of the United States and President of the Senate.

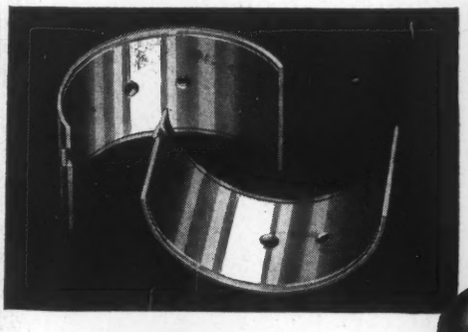
ONCE in a while a great invention is not patentable. Such was the case when Isaac Babbitt in 1839 discovered the remarkable bearing metal which he called "Babbitt." He combined tin, copper and antimony to produce this alloy. Because his idea was recognized as extremely important, and could not be patented, he was rewarded with a gold medal from the Massachusetts Charitable Mechanics Association and an award of $20,000 from the Congress of the United States.

This honor and award were small recompense, indeed, for the millions of dollars his invention has saved industry. No other bearing metal possesses such a low coefficient of friction. When laminated to either bronze or steel it forms the most efficient and satisfactory bearing for certain applications—particularly for all types of internal combustion engines. Today you will find millions of babbitt-lined bearings in use. In giant diesel engines . . . in all types of aircraft . . . in your car or truck . . . they deliver smooth, quiet performance for thousands of miles or hundreds of hours of steady operation.

Manufacturers find Johnson Bronze a capable, experienced, and dependable source of supply for all types of sleeve bearings, including babbitt-lined. Our advice and counsel are available without obligation.

JOHNSON BRONZE COMPANY

525 SOUTH MILL STREET NEW CASTLE, PA.



Automotive Slip-In Bearings with babbitt permanently bonded to strip steel.

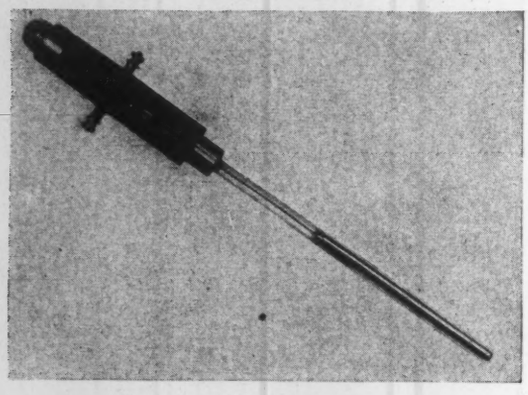
JOHNSON BRONZE
SLEEVE BEARING HEADQUARTERS
BRANCHES IN 18 INDUSTRIAL CENTERS

Bars of Babbitt are used by machine shops in making repairs on the job.

Huge Diesel Bearings are bronze-back, babbitt-lined.

JOHNSON

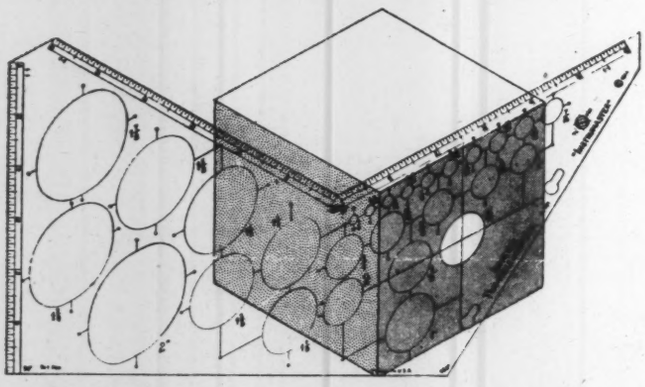
regulator is operating properly. The regulator known as the Mercuroplat, is intended for accurate control of temperatures in water, oil or air baths, etc.; at any temperature from minus 30 F to plus 500 F. It has a sensitivity of



plus or minus 0.02 F. The regulator which is available in lengths from 6 to 12 inches, has machined plastic head, providing visibility for easy reading.

Isometric Stencils

SIMPLIFYING three-dimensional drafting are stencil-type drafting instruments for use in making isometric drawings. Known as Instrumaster, the stencils are manufactured by the Instrumaster Industries, 7357 Arch St., Greenwich, Conn. Drawings may easily be made by using these instruments alone, for the stencils provide inch graduations at full scale along one vertical and two slanted edges. These three edges represent correct directions of



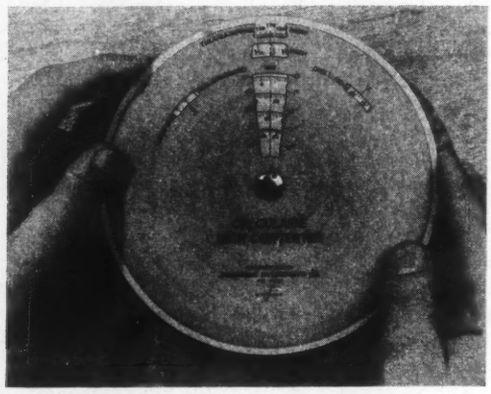
isometric axes. Each stencil is provided with 27 elliptical openings correctly represented in the individual projections, and circles from ⅛ to 2 inches in diameter. By placing the stencil with one edge against a straight edge, the graduated edges as well as the individual elliptical openings automatically assume their required positions within the drawing. The stencils are precision cut of clear high-grade plastic stock 0.040-in. thick, with graduations and lettering impressed in the lower side.

Drafting Stencil

TIME-SAVING, a new drawing board instrument has been announced by RapiDesign, Inc., P. O. Box 592, Glendale, Calif. The instrument which functions on the stencil principle embodies templates for standard gears, rivets, drill points, countersink points, and either Phillips head or Reed and Prince screws. The instrument is made from transparent heavy-gage, laminated-instrument-quality plastic sheets with the markings inlaid between laminations so as not to rub off. Measuring approximately 9 x 12½ inches, it is known as Rapidesign Designer No. 250.

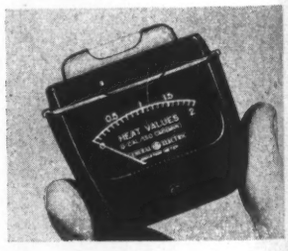
Dimension Converter

FRACTIONS of an inch may be converted into decimals or millimeters, and corresponding U. S. standard gage numbers and drill numbers determined with the aid of the inch-



converter manufactured by the American Hydromath Co., 145 W. 57th Street, New York. Known as Calculaide, the converter is 5¾ inches in diameter and manufactured from tough, nonwarping, noninflammable plastics. Decimal equivalents are graduated in 0.001-inch and can be estimated to an accuracy of 1/5000 of the range.

Radiation Meter



SMALL, portable radiation meter, completely self-contained, has been announced by the General Electric Co., Schenectady 5, N. Y. The receiver is a sensitive thermocouple of the total radiation type, sensitive to wave lengths from 0.3 to 3.5 microns. The indicating instrument is a permanent-magnet, moving-coil millivoltmeter having a range from 0 to 2 gram-calories per square centimeter per minute. Accuracy is ±5 per cent full scale value. The instrument is said to be particularly useful in laboratory tests of the absorption or transmission properties of materials through the visible and into the infra-red spectrum.

New Microfilming Process

COMPLETE UNIT, which processes film on the premises within an hour after negative exposure has been announced by Diebold, Inc., Canton 2, Ohio. Known as Flofilm, three units are available to photograph, process, and read a subject material. The camera, which will make 35-mm

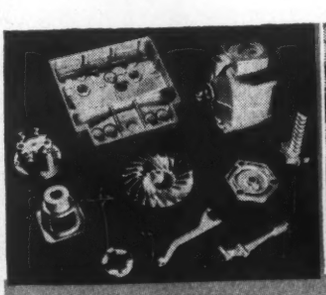
Whatever your needs may be, there's an available magnesium form for every standard purpose. Like other firms throughout the country, you'll find it pays to adapt magnesium's lightness, strength, machinability to the job of cutting costs and improving products in your own field.

Today magnesium alloys are freely available in all common forms—ingots and billets; sand, permanent mold, and die castings; forgings; extruded bars, rods, shapes, and tubing; and rolled sheet, plate, and strip. Commercially pure magnesium and such special products as welding rod, protective agents, fluxes, and special alloys are also readily available. As the pioneer producer and leading fabricator, Dow offers complete technical data and assistance in working and fabricating magnesium. Simply contact the nearest Dow office.

Ready... to make products move!

MAGNESIUM

LIGHTEST OF ALL STRUCTURAL METALS



Die casting magnesium is an accepted method of producing lightweight parts with good surfaces and close dimensional tolerances.

Preheated magnesium sheet is readily deep drawn in a single operation, without need for repeated annealing and redrawing.

Sand and permanent mold castings of magnesium are known for their exceptional combination of lightness, soundness, strength.

MAGNESIUM DIVISION • THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN

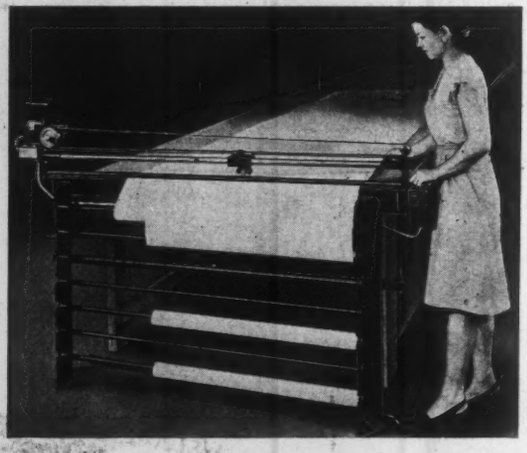
New York • Boston • Philadelphia • Washington • Cleveland • Detroit • Chicago • St. Louis • Houston • San Francisco • Los Angeles • Seattle

or double 16-mm images, will accommodate subjects 14 inches wide and of any length. Resembling a four-drawer letter file, the unit includes a positive printer. The processor is an automatic machine which completely processes exposed film. It will process any length of 35-mm film, can be loaded in three minutes, and its simple and

easy operation may be performed by any office personnel. The reader is an all-electric unit accommodating either 16-mm or 35-mm film. It accommodates rolls, strips or mounted microfilm.

Rapid Paper Cutter

PROVIDING an efficient, economical way of cutting paper roll stocks into sheets of any desired length in one operation, a new Electromatic paper cutter has been an-



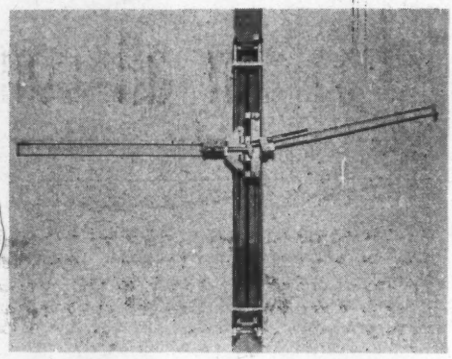
nounced by the Charles Bruning Co., 4754-12 Montrose Ave., Chicago 41, Ill. The unit is designed to mount on one end of a table equipped with scale for measuring sheet length. The cutter is sturdily built and is said to provide a precise way of cutting blueprint paper.

Circular Slide Rule

PRECISION of a 12 inch slide rule is combined with pocket-size convenience in a circular slide rule announced by the American Hydromath Co., 145 West 57th Street, New York. Known as Calculaide Commerce Slide-Rule, it has a C-D scale 12½ inches long, and a log-log scale, ranging from 1.01 to 100, arranged as a three-turn spiral. The instrument is manufactured from tough, nonwarping, nonflammable plastic, and scales are printed in contrasting colors. It is furnished in an attractive leather case.

Stereographic Drafting Machine

PERSPECTIVES and stereographs are easily drawn with the aid of the Pomeroy stereograph drawing machine made by the Pomeroy Stereograph Co., Inc., 1783 E. Eleventh



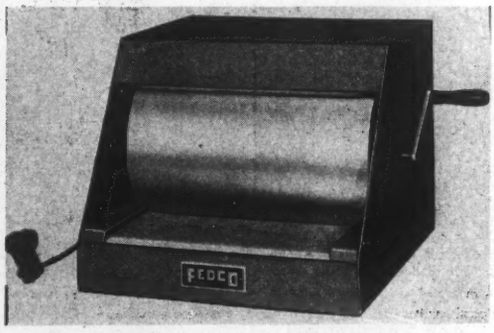
St., Cleveland 14. Three-dimensional views can be drawn for a wide variety of station points both above and below the subject.

Radii Instrument

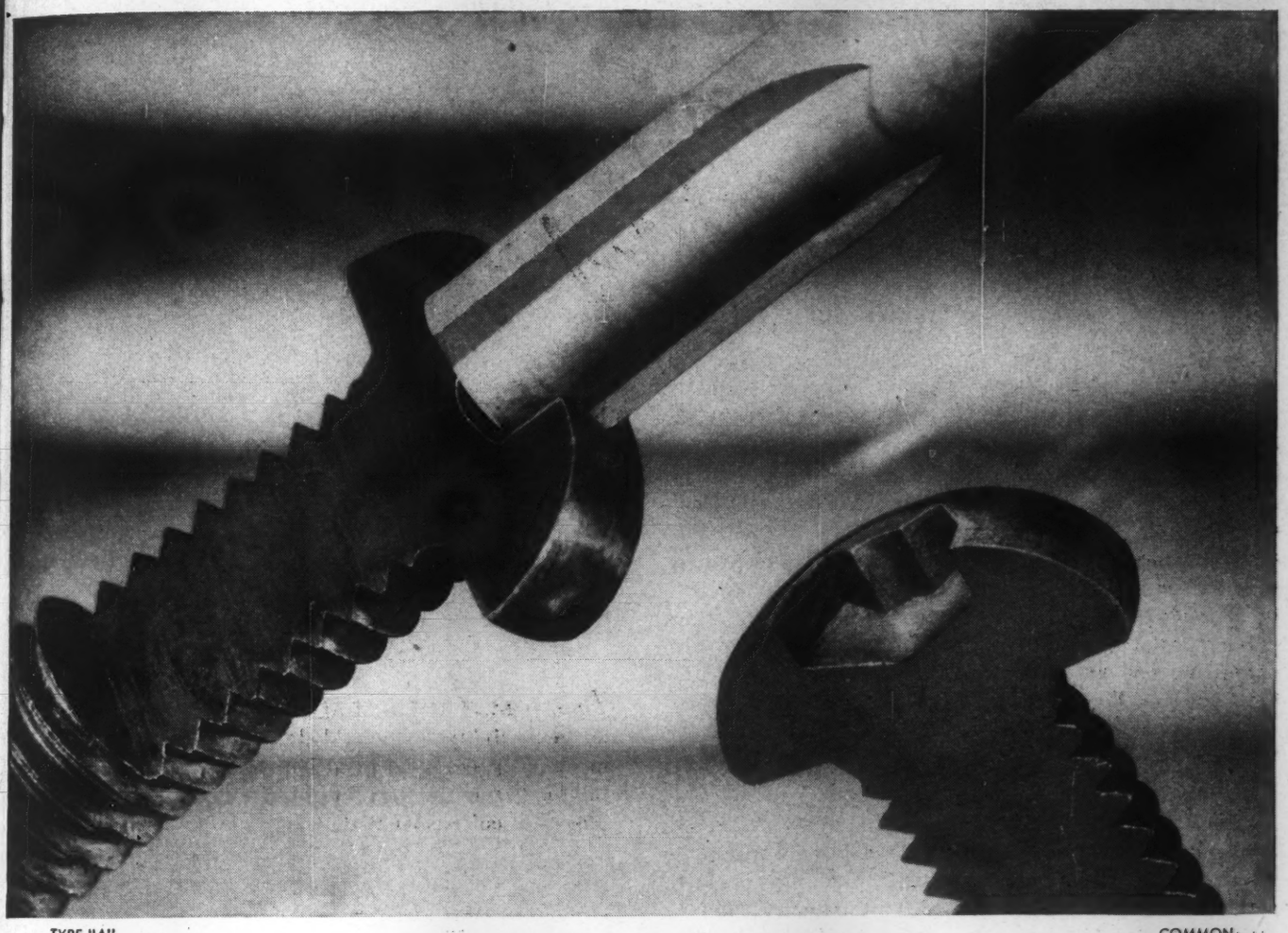
OFFERING to draftsmen a rapid method of determining radii, a template type instrument has been announced by RapiDesign Inc., P. O. Box 592, Glendale, Calif. The instrument is used by merely placing the template tangent to the ends of the lines to be joined. Known as Radius Master, the instrument provides radii up to 37/32-inch in increments of 1/32-inch. The template is made from transparent plastic material.

Print Dryer

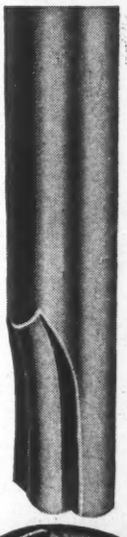
USEFUL for drying photostats, copy prints, and blueprints, a new dryer has been announced by the Fedco Products Company, 37 Murray Street, New York. Available in two sizes, the unit, known as the Fedco Drum Dryer, will handle as many as one hundred and twenty 9 by 11 photostats per hour when motor driven. Occupy-



ing a space 18 inches square, the dryer utilizes a continuous single-belt made of duck and is internally heated by a 300-watt electric element.



TYPE "A" ASSEMBLY BIT



COMMON SCREWDRIVER

THE INSIDE STORY OF CLUTCH HEADS

Here It Is, showing *what* goes on inside the Clutch recess, and *why* . . . why users testify so freely that CLUTCH HEAD has features not matched by any other screw on the market for safety, speed, and savings.

Visibility a Speed Factor. The wide, roomy Clutch invites operator confidence. Saves "breaking-in" period. Presents an easy-to-hit target for stepped up production.

No Driver Canting to chew up heads and create a skid hazard. The Center Pivot Column on the Type "A" Bit guides the driver into the mating recess formation. Deep dead-center entry and straight driving become automatic.

Safe, Effortless Drive-Home. Note straight sides of the driver matching straight walls of the Clutch recess. Contact is all-square on a broad pressure area for positive torque grip and non-tapered driving.

No Ride-Out to Combat. This feature disposes of "kick-out" as set up by tapered driving . . . the commonest cause of driver slippage. The screw rides home easily without the application of fatiguing end pressure.

Lock-On Breaks "Bottlenecks". A reverse turn of the bit in the recess unites screw and driver as a unit for easy one-handed reaching to hard-to-get-at spots. Normal driving of the screw releases the Lock-On.

Sample screws and Type "A" Bit sent on request

"Outlasts Other Bits 5 TO 1"

The rugged structure of this Type "A" Bit logically explains this testimony. Also, a 60-second application of the end surface to a grinding wheel repeatedly *restores it to original efficiency. No expense. No delay.*

CLUTCH HEAD SCREWS

Simplifies Field Service

This is the only modern screw basically designed to operate with an ordinary screwdriver. With a Type "A" hand driver, the Lock-On feature permits the withdrawing of screws undamaged and held safely for re-use.

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

Noteworthy Patents

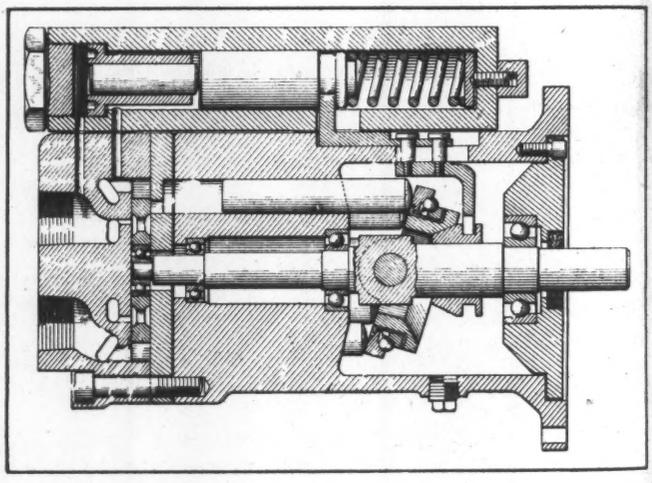
EFFECTIVE RETENTION OF LUBRICANT as well as exclusion of dust, dirt and other foreign matter are afforded by a new design of antifriction screw covered in patent 2,398,789. Unlike previous devices of similar nature, the nut is the longer member and the screw is the shorter member with the antifriction balls being circulated through the screw rather than through the nut. An external recirculating channel is thus dispensed with, simplifying installation problems. Alternate balls used are undersized and act only as spacers. Patent assigned to the Cleveland Pneumatic Tool Co. by Henry S. Hoffar.

DISPENSING WITH LAPPED FITS, a new design for fluid pumps or motors allows the use of overall manufacturing tolerances as wide as 0.003-inch to provide low-cost, quantity production. Covered by patent 2,393,558 assigned to the Acrotorque Co., by Elias Orshansky, the mechanism utilizes a sliding and literally self-aligning piston sleeve to direct flow to and from the piston chamber. A self-aligning sleeve also houses the piston sleeve for simplified interchangeable assembly and all seals are by means of piston rings. Functioning is dependable under wide variations of viscosity, temperature, and pressure. Complete units weigh less per horsepower than present standard equipment and capacity can be varied to suit.

ELIMINATION OF HUNTING characteristics normally encountered with many tracer control mechanisms for machine duplicating is achieved by a device covered in patent 2,395,525. Assigned to Continental Machines, Inc., by James W. Wilkie and Edmond G. Franklin, the tracing mechanism produces duplicates with the continuous, smooth contours of the pattern rather than the usual step-like surfaces created when translation between tracer and cutter lags. Sensitivity is of such an order that even relatively soft pattern materials such as plaster can be used.

INDEPENDENT DIRECTIONAL CONTROL of the fluid motor as well as simultaneous control of the displacement of both the pump and motor is claimed for an unusual hydraulic transmission covered in patent 2,395,980. Assigned to Sunstrand Machine Co., by Gunnar A. Wahlmark, the arrangement utilizes wobble type pump and motor with the wobble plates interconnected. As the displacement of the pump is decreased, the control system acts to increase the displacement of the motor so as to obtain large torques and low speeds for machine feed rates and as pump displacement is increased, operates to decrease motor displacement for low torques with high speeds for rapid traverse operation.

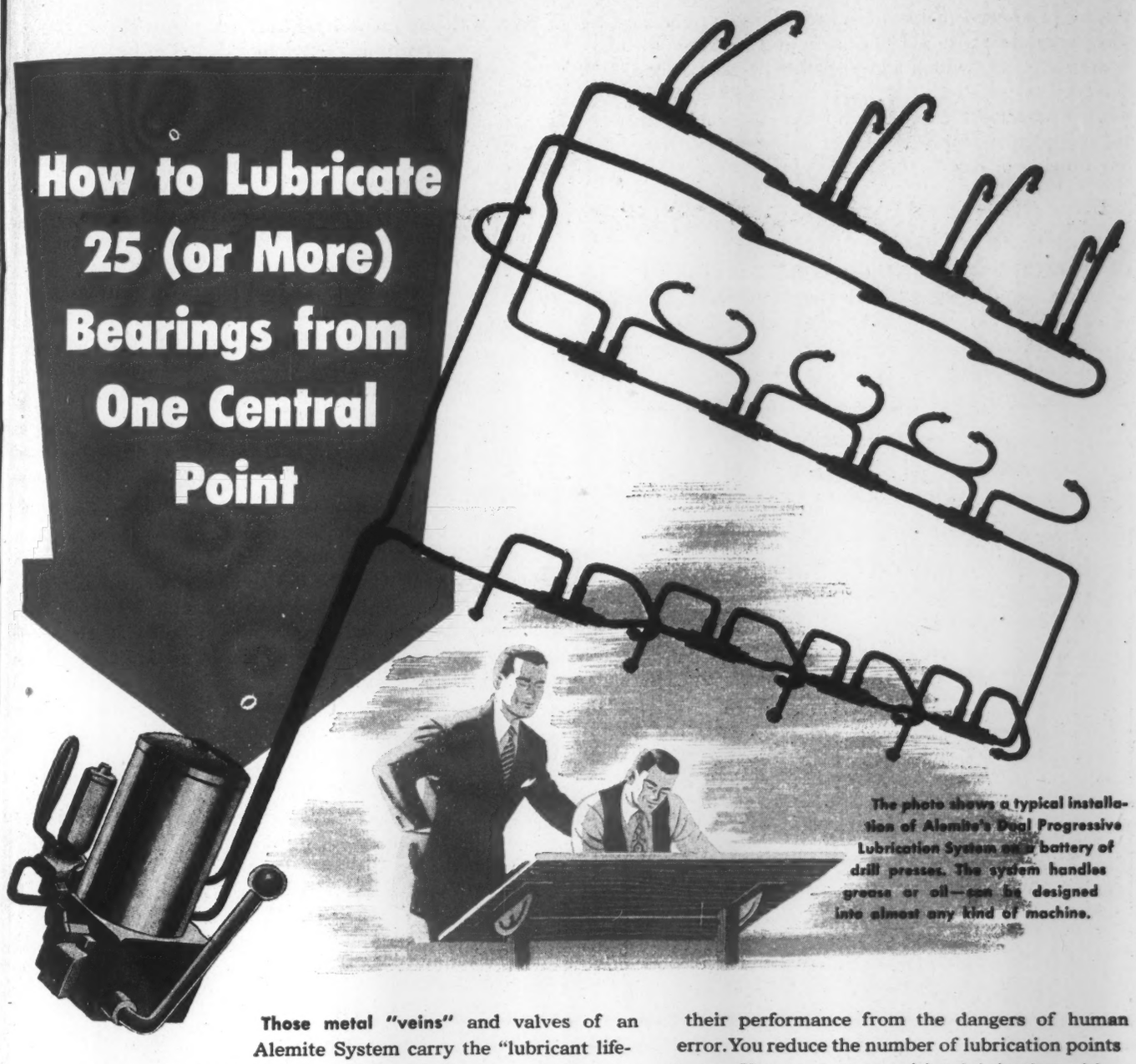
AUTOMATIC DISPLACEMENT CONTROL as well as reversibility is provided by the hydraulic motor or pumping device covered by patent 2,392,980. Assigned to the Crane Co. by Donald G. Fawkes, the unit employs a circular valve plate which gyrates in timed relation



with the wobble-plate driver to direct the flow of fluid properly. A double piston allows for control pressure on the stroke-shifting device regardless of the direction of fluid flow.

AUTOMATIC FREQUENCY REGULATION of alternating-current generators is provided by a continuous speed-control mechanism covered in patent 2,395,707. This control, applied through the medium of a diesel fuel-governing device, utilizes an independently-driven master clock operating in conjunction with a generator powered synchronous motor clock through a planetary differential

How to Lubricate 25 (or More) Bearings from One Central Point



The photo shows a typical installation of Alemite's Dual Progressive Lubrication System on a battery of drill presses. The system handles grease or oil—can be designed into almost any kind of machine.

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Proof? For example: a battery of 7 big automatic punch presses has a total of 86 bearings. It took 20 minutes to lubricate each press—and it had to be shut down. Bearing failures were frequent. Now all 86 bearings are lubricated from one central point *while the presses are in operation.* Bearing failures due to faulty lubrication have ended.

When you design Alemite Centralized Lubrication into your machines you automatically protect their performance from the dangers of human error. You reduce the number of lubrication points to *one.* You guarantee positive lubrication without "time-outs." You cut repairs to a minimum . . . lengthen machine life . . . increase customer satisfaction. It all adds up to More Productive Time per machine. And that enhances your reputation as a designer.

* * *

Without obligation, have an Alemite Lubrication Specialist demonstrate one or all 4 new Centralized Systems right at your desk with transparent working models. Also, ask for any technical help you want regarding Alemite Systems. Drop a note on your letterhead to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

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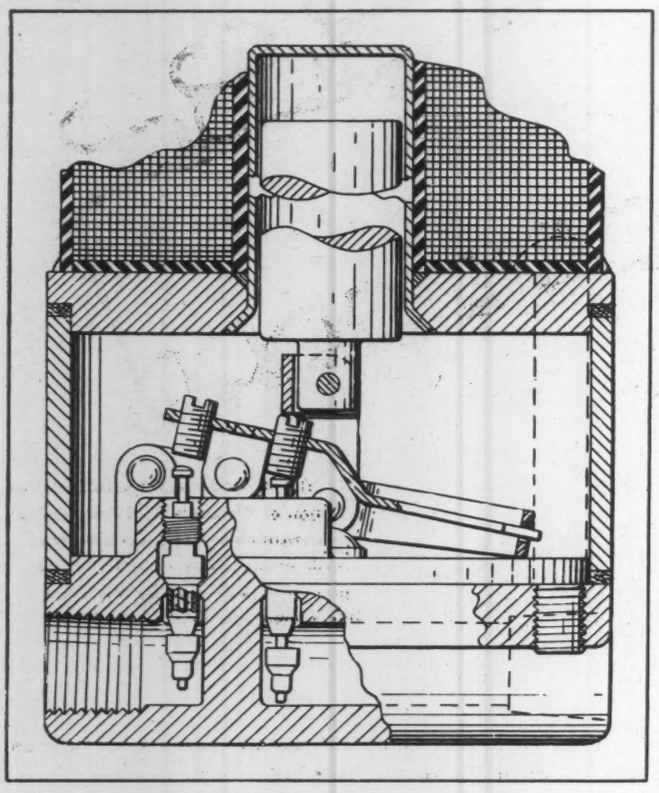
system to control motor speed. Generator frequency is thus controlled within limits sufficiently narrow to hold household clocks within an acceptable degree of accuracy. Assigned to the Ex-Cell-O Corp., by Carroll R. Alden, the basic arrangement covered in the patent can be adapted for other types of prime movers and may be applied to any number of units.

SIMPLICITY OF MANUFACTURE and servicing, outstanding features of a novel 3-way solenoid air valve covered in patent 2,398,452, are provided by utilization of standard automotive inner tube valves. Assigned to



the Penn Electric Switch Co. by Burton E. Shaw, the design provides a pair of such valve units for controlling three-way valving by a solenoid-operated lever system.

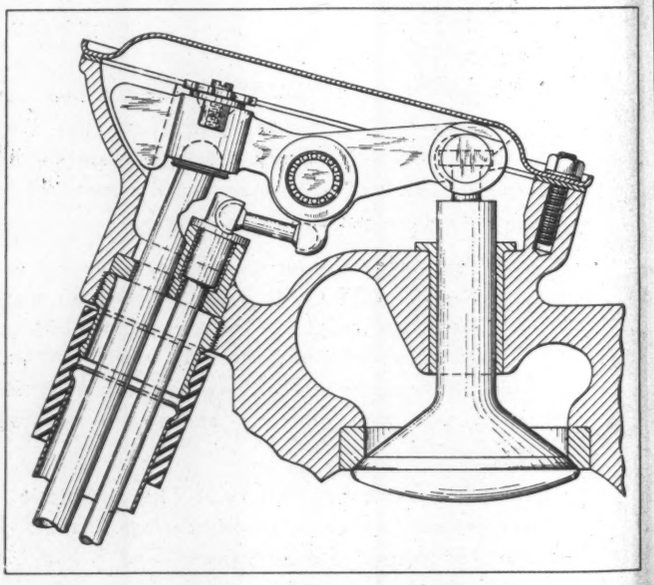
INTERCONVERSION OF RECIPROCATION AND ROTATION is obtained with a mechanism covered by patent 2,384,290 assigned to Rogers Diesel & Aircraft Corp. by Arne Feroy. Primarily for use in engines, pumps, and compressors of the barrel type where pistons are axial, the design simplifies the manufacture of such arrangements and substantially reduces the weight.

ACCURACY OF CONTROL is assured by the clutch arrangement covered in patent 2,393,835 assigned to the Stevenson Engineering Corp. Difficulties often encountered with mechanical controls are obviated by applying clutching pressure direct to the friction disks by hydraulic means.

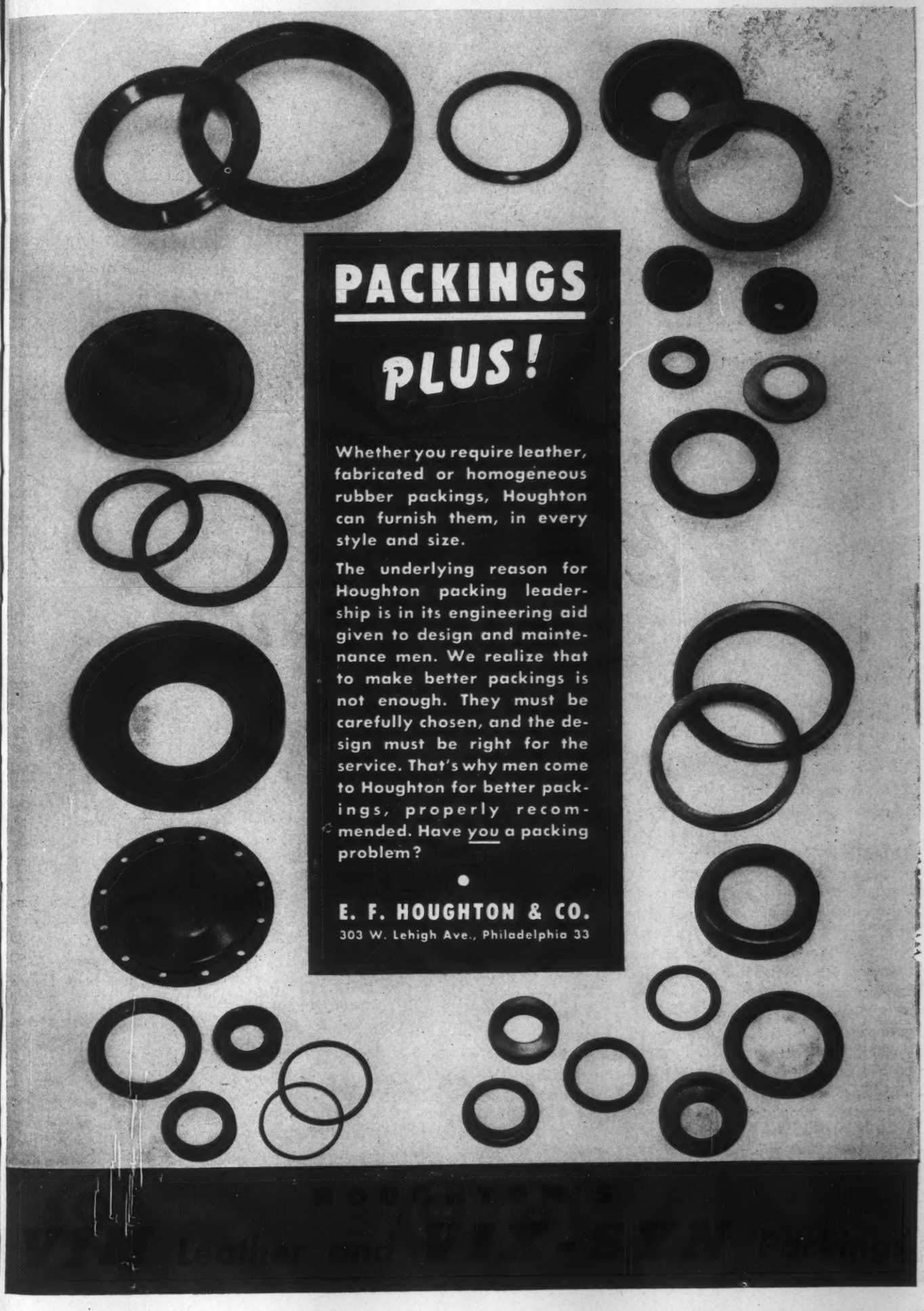
HYDRAULIC BALANCE, permitting better economy, efficiency and long life, is obtained with a novel gear pump and motor design covered in patent 2,399,008 assigned to General Electric Co. by John H. Doran. Utilizing an arrangement of sun and planetary gearing, porting design is such that no resultant forces act upon the sun gear and those forces loading the bearings of the planetary gears are unusually small. A special arrangement of the device can be used as a variable speed coupling or motor.

SERIOUS WEAR of valve heads and seats is prevented by a valve-rotating device assigned to Thompson Products by Eldon K. Ralston under patent 2,397,502. Self-contained and compact, the device effects continuous rotation of valves and similar reciprocating units by small increments. At each valve movement a belleville spring deflects and forces a series of balls by wedge action to rotate the valve stem.

INCREASED COMPACTNESS in overall engine design as well as less valve operating mechanism is made possible by means of a torsion valve spring especially developed for radial aircraft engines. Covered by patent 2,398,207 assigned to the Wright Aeronautical Corp. by



Allan Chilton, the design reduces the height ordinarily required for conventional valve closing means to a bare minimum. The valves used are extremely short and the torsion spring has much less surge inertia than the ordinary helical type.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MEN...of...machines

EUGENE W. O'BRIEN, who has been nominated president of The American Society of Mechanical Engineers, is vice president of the W. R. C. Smith Publishing Co., Atlanta, Ga. Born in West Warwick, R. I., in 1897, Mr O'Brien received his education there in public schools, and was graduated from Brown University in 1919 with the degree of bachelor of science in electrical engineering. He received his master's degree from Brown two years later, and in 1924 was awarded the degree of mchanical engineer by Yale University. Mr. O'Brien was a member of the engineering faculty at Brown until 1922, when he transferred to the faculty of Sheffield Scientific School. He also served as consultant to various New England engineering firms in the fields of mechanical and electrical engineering, power, design and construction. Leaving Sheffield, he joined the engineering firm of Jenks and Ballou, becoming engaged in investigation, design and supervisory work, especially in mechanical and power projects. In 1941 he became a consultant for various government agencies. His connection with the W. R. C. Smith Publishing Co., of which he is now vice president, dates back to 1927 when he was appointed editor-in-chief of Southern Power Journal.

KENNETH N. MACOMBER, associated with Lapointe Machine Tool Co. for the past twelve years, has recently been promoted from chief service engineer to the position of chief engineer. Immediately upon graduating from Northeastern university, he started working in the shop for the Lapointe company, later serving as head of service and sales in the Detroit area. Among his assignments are included the setting up and testing of the 37, 75 and 90 mm breech ring and breech block broaching machines at the Watervliet Arsenal, the first machines used on this type of work during World War II. Similar work was performed by Mr. Macomber for Chrysler, Chain Belt Co., Goodyear, Oldsmobile and Pontiac on 40, 76-mm and 4.7-in. guns. In addition, he had charge of setting up and testing of the propeller spider and propeller cam and the spring pack retainer at Nash-Kelvinator Corp. Mr. Macomber in addition to his position as chief engineer will continue lecturing on broaching, which he has done for the company for the past eight years.

ARTHUR W. CARPENTER has recently been elected president of the American Society for Testing Materials. Mr. Carpenter, who is manager of the testing laboratories, The B. F. Goodrich Co., has been active in the affairs of the

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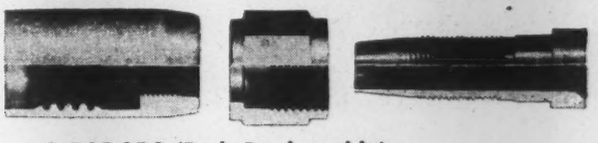

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society since 1928, at which time he acted as secretary of the society's Committee D-11 on Rubber Products. He has taken active part in the society's new work on ultimate consumer goods and is serving as a member of this Administrative Committee. He has also served on the Administrative Committee on Papers and Publications and is vice chairman of the Cleveland District Committee. From 1931 to 1933, and again from 1941 to 1943, he was a member of the executive committee. Prior to his present appointment he completed a term as the society's vice president. Mr. Carpenter received his bachelor of science degree in chemical engineering from the Massachusetts Institute of Technology in 1913, and his master's in 1914. Before going overseas during World War I he was city chemist at Alliance, O., and chemist of Akron Municipal Water Purification Plant. Upon returning from the service, he became affiliated with The Goodyear Tire and Rubber Co., and later with Holtite Mfg. Co. He again joined Goodyear where he remained until 1926, at which time he became associated with B. F. Goodrich Co. Since 1928 he has been manager of the testing laboratories at Goodrich. During World War II he was for many months a consultant in the WPB Conservation Division.

♦

HAROLD T. YOUNGREN, formerly director of engineering for Borg-Warner Corp., has been named director of engineering of Ford Motor Co. R. H. MCCARROLL, an executive engineer at Ford, was promoted to director of chemical and metallurgical engineering and research for the company.

♦

WITOLD S. GWIZDOWSKI, a mechanical engineer formerly with the Cleveland laboratory of NACA, has joined Colonial Broach Co., Detroit, in the same capacity.

♦

ROBERT MORRISON BERNS has resigned as president and chief engineer of Aircraft Marine Engineering Corp., to become partner, designer and manager of The Boatbuilders, Chicago.

♦

LAWRENCE M. ROSE has left his position as mechanical engineer with NACA, Cleveland laboratory, to join General Electric Co., Schenectady, in a similar capacity.

♦

ALVAN MACAULEY, who had served eighteen years as president of the Automobile Manufacturers Association until he resigned that post last February, was awarded the AMA Charles Clifton Award for being "an inspiring leader in achieving automotive teamwork in the production of war materials in World War II. . . ." Mr. Macauley is chairman of the board of Packard Motor Car Co.

Lubricants

(*Concluded from Page* 136)

that these lubricants are corrosive and that their use must be confined to certain specific operations. This theory has retarded a more widespread use of extreme-pressure lubricants throughout industry, preventing the advantages and economy that may be realized from their use. Their development was a valuable contribution by the petroleum refiners, and they are in use by some of our major industries in large circulating systems where as much as ten per cent of water per week finds its way in the system. No corrosion has been experienced with roller bearings which constitute a part of the equipment being lubricated.

Water tends to form a permanent emulsion when mixed with extreme-pressure lubricants which cannot be removed by centrifuging. The discovery of an emulsion breaker which will provide complete separation of the water from the lubricant now makes possible their more general use throughout industry. Use of extreme-pressure lubricants for hypoid gears supported by antifriction bearings in our millions of automobiles, trucks, buses, and tractors for many years is ample evidence of satisfactory performance. Extended use of extreme-pressure lubricants may be expected for worm-gear drives, speed reducers, heavy reduction gear drives, and bearings, etc., as the science of improved lubrication is advanced.

Grease Lubricating Antifriction Bearings

Where machines are designed for grease lubricating antifriction bearings, provisions for flushing and cleaning the bearing by means of a removable plug in the bottom of the bearing housing should be made.

Over-lubrication (too much grease in an antifriction bearing) is perhaps responsible for more lubricating difficulties or bearing failures than any other cause. The best practice is to provide a removable plug in the top and bottom of the bearing housing so that it may be periodically flushed and cleaned with a suitable solvent while the bearing is in motion, removing all of the old, oxidized grease. The bearing should be drained completely, the plug in the bottom of the housing replaced, and the bearing filled not more than one-fourth to one-third full of the proper grease. No additional grease should be added to the bearing between flushing periods. Time elapsed between flushing and relubricating periods will vary slightly from one operation to the other, depending on the severity of service but, by careful supervision on the part of the operator, such periods can be determined and a schedule set up for relubrication. Many progressive operators following this practice find that from six months to one year or more is sufficiently frequent to insure the best lubrication.

If industry is to enjoy efficiency resulting from the maximum in lubrication, we must start with the designer who should have a good working knowledge of lubricants and their proper application of purification equipment to maintain the oil in its original, clean condition, and of the catalytic effects of metals on oils. He should follow the rule of simplicity, minimize the number of oils or greases for a given unit, and provide enclosed systems with adequate seals to minimize contamination.

Then, the lubricant supplier must co-operate by supplying the best and most economical lubricant for a given condition. Finally, the operator or plant lubrication engineer should have a basic knowledge of all factors involved and co-operate by establishing a supervised program to see that the proper lubricants are used, to check the condition of the lubricants periodically and to establish oil change periods in accordance with the best operating practices. Any one or more of the basic principles thrown out of balance can defeat the goal of the utmost in lubricating efficiency.

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Rubber Torsion Springs

(*Concluded from Page* 149)

But $R_1^2 S_1 = T/2\pi L$, from Equation 2, therefore

$$\psi = \frac{T}{4\pi LG}\left(\frac{1}{R_1^2} - \frac{1}{R^2}\right) \quad \ldots\ldots (5)$$

The difficulty in applying such an analysis is that any equation involves the shear modulus G for the rubber compound. It is common practice to assume that G is defined by the slope of the tangent to the shear curve for small amplitudes of movement. Such a tangent is drawn on the shear curve, *Fig.* 5, and departs seriously from the slope of the curve in the range of actual stress.

Since every particle of rubber in the torsion spring may be thought of as an element in the thin layer *dr*, the entire body of rubber is stressed in shear, the shearing stress varying inversely as the square of the radius to the particle (see Equation 2). The foregoing analysis can be used in a graphical approximation which is fairly accurate. This has been done in *Fig.* 5 by assuming that the body of rubber is divided into successive layers each having a thickness equal to 5 per cent of its inside radius. The heavy spiral line represents the position taken under torsional deflection by a theoretical element in the rubber which is radial at no load. In each of the successive layers the distorted element departs from the radial direction by the angle of departure for that stress as obtained on the shear "sandwich".

One such graph is all that is needed for each compound to cover the widest assumed range of unit static stress and torsional rate. For lower static stresses it is only necessary to move out to a new radius where the required shear stress exists and strike a new arc to represent the inside diameter of the rubber.

Controlling Angular Deflection

The amount of angular deflection is controlled by the shearing stress on the rubber and by the thickness of the rubber wall. Soft rubber will deflect more than harder compounds, the range of the most suitable compounds being generally from 45 to 55 durometer; the type of service usually determines the best compound and it is a safe assumption that most calculations should be made for 45 durometer if natural rubber, and 50 durometer if synthetic.

For calculating, it is convenient to measure the thickness of the rubber wall as a percentage of the diameter of the inner member. For instance, if the inner member is a one-inch diameter shaft or tube and the rubber wall one-quarter inch in radial thickness, the wall is twenty-five per cent of the shaft diameter. It is fundamental that all springs having the same per cent wall, if rotated through the same angle, will develop the same shearing stress on the bond between the rubber and the shaft, as indicated by Equation 4.

In the preliminary calculation of a rubber torsion spring, if the elastic deflection required is very great, it will usually be desirable to employ a fairly long arm to avoid excessive angularity. After deciding on some convenient length of arm, the static torque—torque on the spring when the mechanism is at rest—should be calculated. The allowable shear stress between the inner member of the spring and the rubber will depend on the nature of the service. For passenger car suspension springs 120 psi with full load is a satisfactory stress to start with. For mountings of machinery, the allowable stress may be less than half of this.

After determining the static torque and the static stress on the rubber, the diameter of the inner member of the spring, and its length, may be fixed at any combination which will give the estimated shear at the required torque. The decision as to the combination of length and diameter will depend on the elasticity of flexibility (other than angular) desired.

Determining Proper Thickness of Rubber

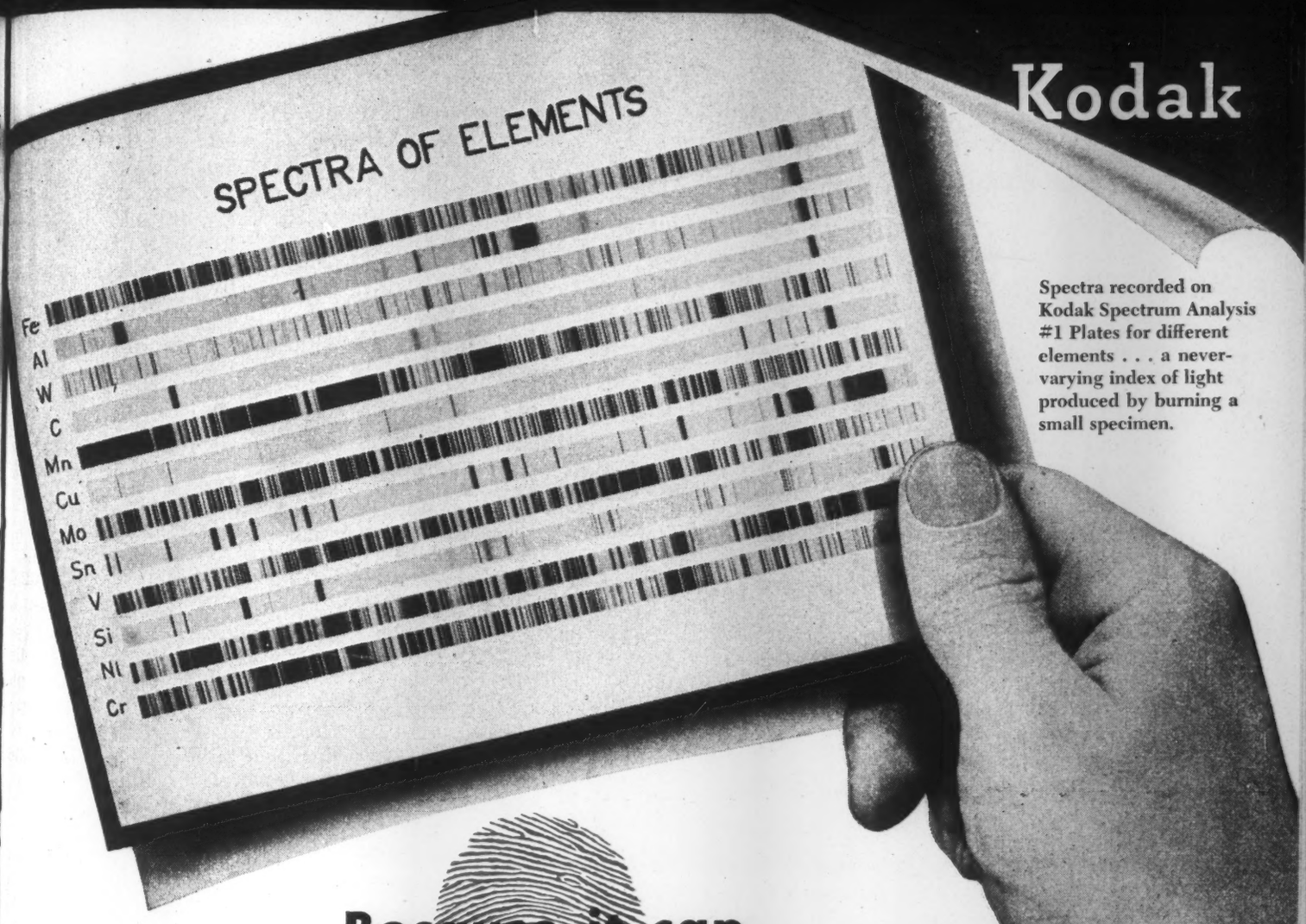
The only remaining dimension to calculate is the radial thickness of the wall of rubber. To obtain this, it is usually easiest to calculate either the angular rate of the spring or the angular wind-up in static position. Static wind-up will usually be less than 45 degrees. The graph of spring characteristics, *Fig.* 6, shows the relation between wind-up stress for rubber wall thickness varying from 10 to 50 per cent of the shaft diameter. The graph is made for 42 durometer stock. If 50 durometer is to be used, the stress for a given angle should be multiplied by 1.37, and by 1.62 for 55 durometer.

Maximum permissible shear stress on the bond between the rubber and the shaft depends on many factors such as frequency at which the maximum occurs, and stress range. For unusual or infrequent load application, which occur seldom in the life of the spring, extreme stresses as high as 300 psi have been successfully used. For most applications the maximum should be lower than this.

The range through which the stress on the rubber varies will determine to some extent the life expectancy of the spring. If the spring is flexed continually from free condition to 300 psi, the life will be relatively short. If the range is from 120 to 130 psi, the life will be longer than if it is from zero to 100 psi. Reversing the direction of stress, as from plus 50 psi to minus 50 psi, will give shorter life than where the stress is always in the same direction.

In general, best results can be obtained where there is no eccentric loading, that is, where the torque arm applies the load near the center of the spring length, although this is not always possible. If the spring is relatively long, as for instance five or more times the shaft diameter, it will resist eccentric loading or tilting much better than a short spring. A spring with a relatively thin wall of rubber—less than 20 per cent of the shaft diameter—will resist tilting better than a thick-walled spring.

The rubber torsion spring performs its own locating functions; that is, it resists axial movement and tilt. The slight flexibility in these directions is a distinct advantage in most installations as it absorbs impacts in every direction. This also makes unnecessary the added cost and weight of bearings and bearing seals and eliminates squeaks and rattles and maintenance operations such as lubrication. Rubber torsion springs designed within the limits outlined here should be capable of many years use without any further attention or service other than a possible adjustment at long intervals.



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19

Roller Chains

(*Concluded from Page* 152)

any provision for pitch-wear take-up. The chains were installed with an initial tension of about 25 per cent of the working load on both strands. Under this condition the chain drive has practically no backlash, and no appreciable looseness developed after prolonged periods of operation. Moreover, the drives set up with this minimum amount of slack were quiet and smooth in operation from the start. No "break-in" period is required for marine chain drives, and full sustained loads may be applied immediately.

It will be noted that separate chains are used from each engine to the main sprocket, *Figs.* 2, 3 and 4. With this system maximum wrap is obtained without idlers, and the drive retains the desirable feature of ability to operate with one engine in case of accident to one of the chains.

Sprockets in marine chain transmissions have almost unlimited life. Steel of machinable hardness or high-strength cast iron often is used. As there is no rubbing action on the sprocket teeth, and as the driving forces are distributed over a relatively large number of teeth, very little sprocket wear occurs.

Lubrication Not a Serious Problem

LUBRICATION: Chain drives are not critical as regards oil viscosity. Any medium oil which is fluid enough at all operating temperatures to reach the joint or "heart" of the chain is suitable. The type of oil used for the engine lube system seems to be ideal for the marine chain drive, and the use of engine lubricating oil in the chain housing has been standard practice in nearly all installations. The oil, however, should never be taken from the engine system. A separate sump, strainer, and pump should be provided for the chain transmission, preferably driven by a gear or chain from one of the pinion-sprocket shafts. Spray pipes should be placed to distribute the oil to all portions of the chain. Cooling of oil is not always necessary. When chain speed is less than 2500 fpm, oil temperatures seldom exceed 130 F.

HOUSINGS: The drives were constructed at a time when horizontal-boring-mill facilities in the country were critically needed for work on helical gear reducers and other large machinery. For this reason the transmissions were designed to utilize standard pillow blocks with self-aligning roller bearings. The bases were weldments of 1-inch plate well ribbed to assure stability. The sheet-steel tops functioned only as enclosures.

This arrangement worked out well and can be recommended as lending itself to economical manufacture especially where the drives must be custom-built singly or in small lots. Thrust bearings of either the roller or slipper type may be incorporated into the transmission. The axial movement of the propeller shaft resulting with the use of the Kingsbury-type bearing is easily accommodated with the chain drive because of the generous clearances between the sprocket faces and the chain side bars.

EQUALIZATION OF ENGINE OUTPUT: Inasmuch as the engines are locked in rotational step, it is easy to expect that there might be some trouble in keeping the engines in good power balance, especially on craft like tugs where frequent starting, stopping and speed changes are necessary. However, most of the troubles which had been anticipated did not materialize.

Pilot-house adjustment of speed has been successful with both pneumatic and hydraulic throttle-control systems. Once the usual preliminary adjustments have been made, the two engines remain in good torque relationship over the entire speed range. Occasionally the exhaust temperatures are checked and slight adjustments of the governors made by the operator. At no condition of operation has there been noticed a tendency of one engine to "fight" or oppose the other. During trial runs deliberate unbalancing of the engines seemed to result in nothing other than a dropping off in speed of the system.

Compensating for Inelastic Drive

HYDRAULIC COUPLINGS: Chain drives undoubtedly respond to the smoothing-out action of hydraulic or magnetic couplings fully as much as would a gear drive under the same conditions. As pointed out previously, the chain is not to any degree elastic nor does it possess damping ability; consequently, abnormal engine cyclic variations or vibrations must be met either by increasing the size of the chain or by introducing an absorbing device between the engine and the drive.

Hydraulic couplings used on these tugs were of the traction type, with the outer and heavier elements supported by the chain-drive bearings. Scoop-type hydraulic couplings or magnetic couplings may be used, making it possible easily to disengage one of the engines for light operating conditions or in an emergency. Whether or not a reaction-type coupling is used, it is advisable to incorporate some form of flexible coupling between the engine and the load.

A coupling which allows easy disconnection is recommended when other means are not available to disengage one of the engines.

Chain Drive Allows Adequate Spacing

Many of the advantages of using two or more lightweight medium-speed diesels with a single propeller may be inherent in a gear-driven multiple-engined vessel. It will often be found, however, that gear drives of suitable ratio, unless constructed with expensive, inefficient idlers, will require that the engines be placed so close together that proper provision for operation and maintenance cannot be made.

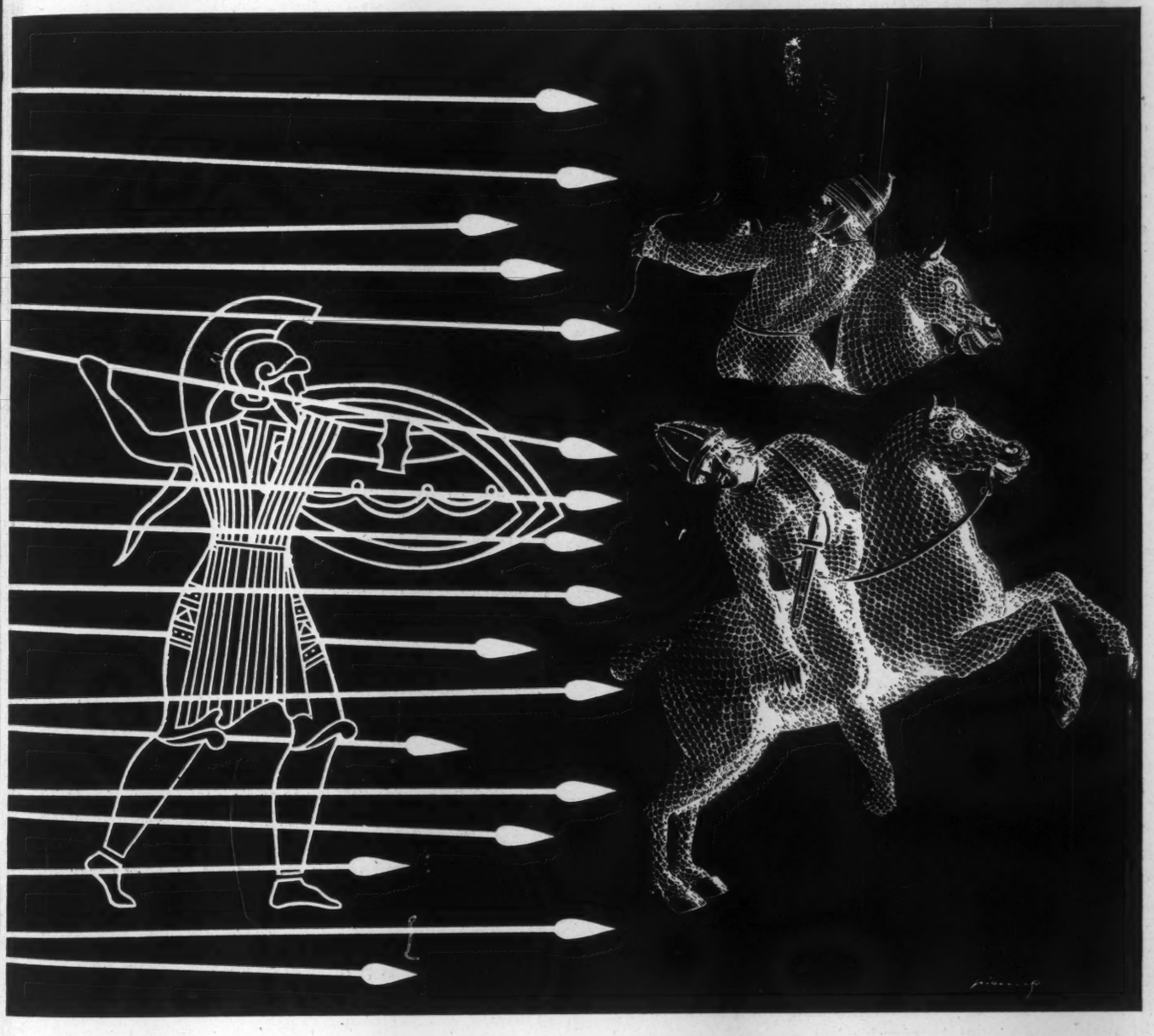
Chain drives, on the other hand, operate best with the sprocket centers spaced at a distance roughly equivalent to one or two times the diameter of the largest sprocket. This relatively wide spacing of centers allows engine spacing in the ship which will provide the best accessibility, balance and arrangement of accessory equipment. Also, because of their less exacting requirements for accuracy of center distance and housing structural rigidity, chain drives lend themselves better than gear drives to economical "tailor-made" design and construction.

Equally as important as accurate chains and sprockets are a base or housing of good design, adequate bearings and shafting size, and fitting and assembly workmanship consistent with the requirements of heavy-duty equipment.

THE MACEDONIAN SURPRISE PARTY

When the proud Persian hordes plunged headlong at Philip of Macedon's army, they were dumped into the minor leagues by an entirely new strategy, the phalanx: a solid wall of warriors sixteen ranks deep. Strength-in-depth withstood and defeated the impact of an over-confident enemy.

Molybdenum steels are economical means of getting the strength-in-depth called hardenability. With it, you're assured of dependable performance under severe service conditions. Practical facts are available to show you where molybdenum can go to work for you.



MOLYBDIC OXIDE—BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"
CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

Climax Molybdenum Company
500 Fifth Avenue • New York City

MOLY

ASSETS to a BOOKCASE

Principles of Physics

By F. W. Sears, Professor of Physics, Massachusetts Institute of Technology, published by the Addison-Wesley Press Inc., Cambridge, Mass.; 400 pages, 6½ x 9¼ in., clothbound; available through MACHINE DESIGN, *$5.00 postpaid.*

As is being belatedly recognized by the educator, it is of paramount importance that the designer be well grounded in the fundamental principles on which his designs are based. In consideration of this, great emphasis is being given to instruction in physics in undergraduate schools. There remains, however, the necessity of periodically reacquainting the designer with the premises from which he works. Recommended to the designer as a reference book rather than a text, this work will form a ready source of information on the principles of instruments with which he must deal, and on the derivation of equations concerning phenomena occurring in his work. Electricity and magnetism are covered in this second volume in a series by the author dealing with physical principles. It treats of such basic fundamentals as: Systems of electrical units, capacitance, inductance, dielectrics, direct and alternating current circuits, electrical instruments, and electronics.

□ □ □

Waveform Analysis

By R. G. Manley, vibration engineer, Silentbloc, Ltd.; published by John Wiley and Sons, New York; 275 pages, 5⅝ by 8⅝-inches, clothbound; available through MACHINE DESIGN, *$4.00 postpaid.*

Vibration recording machines have been developed to a considerably advanced state, and the use of such instruments as the oscilloscope-camera combination is becoming well understood. There remains, however, analysis of the record so produced, in such a manner as to remove spurious and perhaps carrier traces. This book was written to supply practical information on the interpretation of recorded waveforms and allied subjects. Included are sufficient details of basic theory and methods of analysis to enable the reader to acquire the essential groundwork of knowledge concerning the properties of complex waveforms, and to pursue the study of any particular method of analysis to a moderately advanced stage. A feature of the book is a detailed presentation of the envelope method of analysis. This makes it possible for recorded waveforms to be analyzed into their principal constituents without recourse to mechanical contrivances or the more exact but cumbersome methods such as "Harmonic analysis". Throughout the volume particular consideration is given to the needs of scientists and engineers who may be required to interpret quantities of recorded waveforms by analysis.

□ □ □

The Engineer in Society

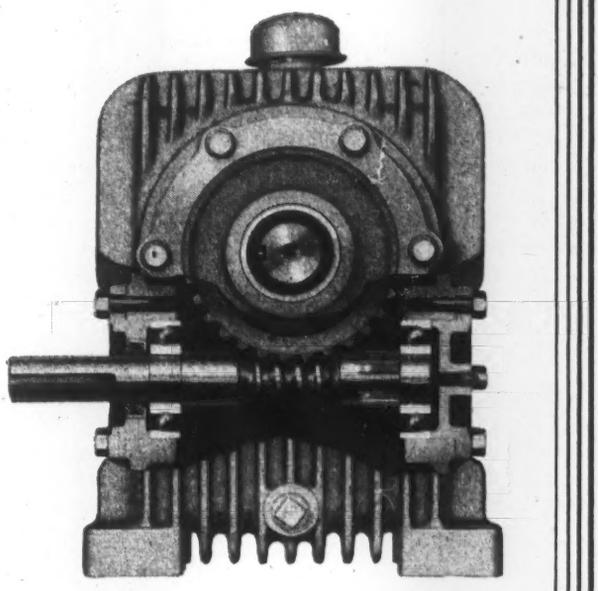
By John Mills, retired Director of Publications, Bell Telephone Laboratories; published by D. Van Nostrand, New York; 196 pages, 5¾ by 8¼-in., clothbound; available through MACHINE DESIGN, *$2.50 postpaid.*

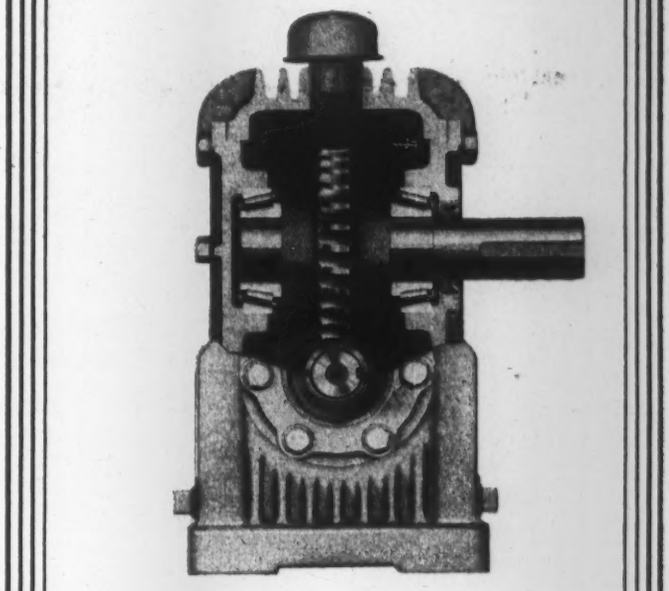
That the engineer is economically and publicly unrecognized, is a view shared by many in the engineering profession. Acknowledging this fact, the author presents in a thoughtful, readable manner his concept of the conditions, causes and cures. The technique which has come to be known as the "scientific method" has aided science in its advancement to such a degree that sociology and the humanities have been left far behind. But, as the author points out, scientific objectivity may well be the solution to the current economic problems. He presents his views on the salaries of engineers, woman's place in the technical field, the use of aptitude tests, and the organization of a professional group to promote the welfare of scientists and engineers.

Published recently by Culco Engineering, a highly interesting paperbound booklet, "Handbook of Dry Plate Rectifier Applications", presents to the engineer probably what is one of the first attempts to cover the application and use of dry plate rectifiers. Written by G. A. Culbertson, consultant on dry plate rectifiers, this first edition of the handbook is based on some twenty years of field experience and covers power applications, cooling, circuit characteristics, and relays and circuit breaker control.

The booklet is well written and should be of considerable assistance to those considering the application of selenium or other dry plate rectifiers where capacities required may run as high as 40,000 amperes. It is available at $0.50 prepaid from Culco, Box 122, Montrose, Calif.

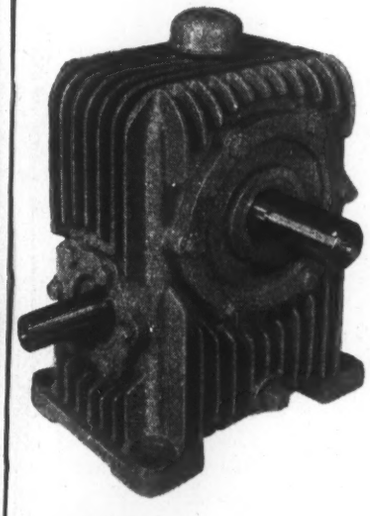
Pertiment data concerning specification tolerances, and size availability, are available in the new booklet "Steel Products Manual." Concerned with stainless and heat-resisting steels, the 152-page booklet covers a variety of production forms such as strip, bar, and wire. A well indexed volume with an excellent glossary, it is available at $0.25 postpaid from the American Iron & Steel Institute, 350 Fifth Avenue, New York, N. Y.





Announcing

STANDARDIZED CONE-DRIVE SPEED REDUCERS



Smaller center distances, made possible by double-enveloping Cone-Drive gearing with its greater contact area per tooth and more teeth in contact mean remarkable compactness and light weight despite their heavy duty construction. All standard models carried in stock to suit practically every requirement—from fractional hp. to 800 hp. input to the pinion.

Models include—pinion under, pinion over, gear shaft vertical or horizontal. Ratios 5:1 to 70:1. Extended pinion and gearshaft models available. Cooling fins provided on all standard models, greatly increasing thermal capacity. Larger models designed for water cooling where required.

Ask for Bulletin No. 789

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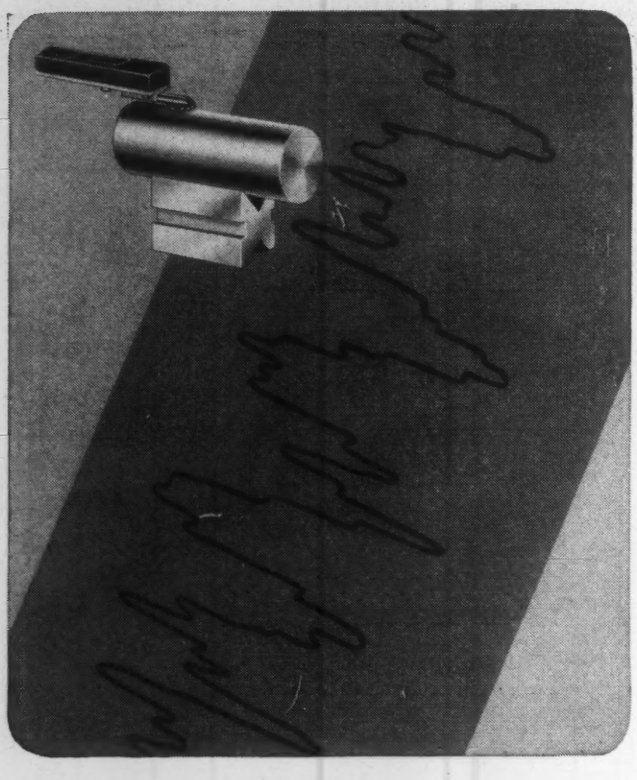

Is your Production Curve Linked to Surface Finish Control?

The Surface Analyzer* checks surface finish from less than 1 to 5000 microinches! Permanent, instantaneous direct-inking oscillograph chart record is obtained by exploring surface with diamond point — the motion of which is magnified and recorded on a moving paper chart. In cases where "peak and valley" chart profiles are not needed, the Brush "RMS" meter provides a constant visual check of average surface roughness.

Human error is eliminated by accurate *power-driven* pickup, chart record and calibration in terms of an absolute standard.

Accessory pickups available for "hard to get at" surfaces such as: small bore measurement, flank, threads and thread gauge applications; tool-edge and wire surface analysis; slot measurement, etc.

For your production needs investigate

The Surface Analyzer

a new aid to industry in surface finish control.

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BUSINESS AND SALES BRIEFS

RECENT additions to its field engineering staff have been made by The Lincoln Electric Co., Cleveland. These are: Gordon Appleby, welding engineer at the Philadelphia branch office: Terril S. Hoffman, welding engineer in the St. Louis area; E. James Langhurst, welding engineer in the Chicago territory; and Garret S. Parsons, welding engineer in the Los Angeles branch office.

♦

Formerly president of Radio Essentials Inc. of Mount Vernon, N. Y., Herman H. Smith has resigned to form his own organization, Herman H. Smith Inc. The new company will manufacture a line of radio and electronic components and hardware, and will act as supplier to radio parts jobbers.

♦

Production has been started in General Electric company's five-million dollar factory at Coshocton, O., which is designed to be the largest and most modern plastics laminating plant in the country. While still under construction, the new plant is replacing present company facilities for the manufacture of laminated materials at Lynn, Mass.

♦

Kimble Glass Co. of Vineland, N. J., has been acquired by Owens-Illinois Glass Co. and will be operated as a wholly owned subsidiary. The present management remains in charge.

♦

After five years in the Army, A. C. Brown Jr. has returned to Air Reduction Sales Co. and will serve as manager of the Cleveland district. Prior to entering the service he was salesman and later branch manager at Grand Rapids, Mich. In his new position he has succeeded Stephen H. Newborn, recently appointed Detroit district manager, who in turn has succeeded G. F. Dekker, newly elected vice president of The Ohio Chemical & Mfg. Co., an Air Reduction subsidiary.

♦

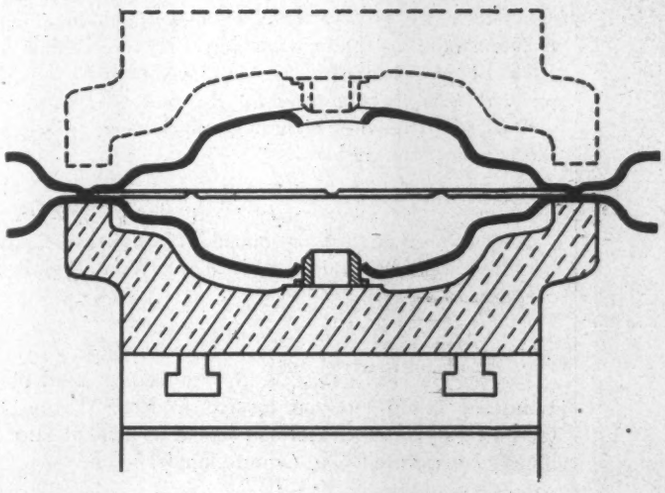
Change of name to Hardy Plastics & Chemical Corp. has been announced by Style Molders Co., 248 McKibbin St., Brooklyn, N. Y.

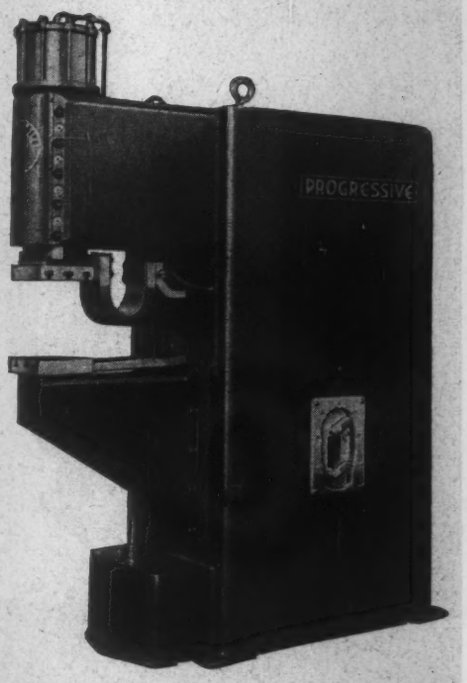
♦

Lovejoy Flexible Coupling Co., Chicago, recently acquired the manufacture and sales of the mechanical power transmission department of Ideal Industries, Sycamore, Ill. Lovejoy now will handle Ideal variable-speed pulleys, adjustable motor bases, "Select-O-Speed" transmissions, Ideal drive sheaves, and stock of wide V-belts.

♦

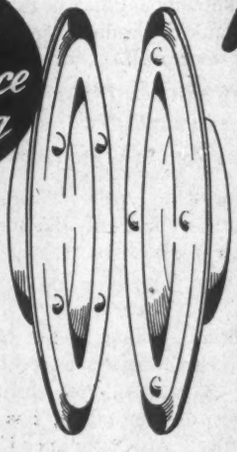
With over twenty years' experience in the carbide and cast alloy industry, Joseph N. Peters has been appointed manager of the Carbide & Cast Alloy Division of Jessop Steel Co., Washington, Pa. He will supervise the production and sale





We cut 75% off his fabricating cost

DESIGNED for Resistance Welding



Top, left... Simple ring dies are used on a Standard Progressive Projection Welder (Top, right) for fabricating steel wheels, out of two identical stampings, each with 4 projections. 8 welds are made simultaneously.

A manufacturer in Illinois was setting up to fabricate a line of toy wagons as a post-war product. He figured they could save money by making each wheel out of two identical stampings spot welded together in 8 places just below the rim.

He was right that spot welding *would* save him money and make a stronger wheel—but he did not know how much MORE he could save by a small change in the stampings.

When Progressive's engineers looked over the design they recommended that each wheel-half be provided with 4 small projections (formed during stamping—see sketch). The two halves could then be dropped into a simple fixture on a standard PROGRESSIVE PROJECTION WELDER—with the projections on one stamping half-way between the projections on the other. With this design, all eight "spots" are projection-welded simultaneously.

This simple design change cut welding and handling time down from over 11 to only 3 seconds per wheel. Desired output of 1200 wheels per hour could now be obtained with one man and one Projection Welder instead of 4 men operating 4 Rocker-Arm Spot Welders.

PROGRESSIVE engineers will be glad to save you money, too, by studying your design for lower cost resistance-welding.

Have you seen the "Idea Book" of 100 typical resistance welding applications. Ask for Booklet No. WP-44.

IT PAYS TO WELD

PROGRESSIVE Welder Co. 3050 E. OUTER DRIVE • DETROIT 12

RESISTANCE WELDING EQUIPMENT

the "Key"

TO LONG DEPENDABLE PERFORMANCE

The successful operation of any gasoline powered appliance, farm machine or industrial equipment depends on good engine performance day after day. The proved performance record of over 2½ Million Briggs & Stratton *air-cooled* engines, built during the past 26 years, is the reason why users, dealers and manufacturers have made Briggs & Stratton powered equipment their *first choice* — and have established Briggs & Stratton engines everywhere as the Right Power for hundreds of applications.

Be assured of years of dependable, trouble-free performance by waiting until you can have equipment powered by Briggs & Stratton engines. You will not have to wait very long — because deliveries are improving with the increased production now flowing out of our factories.

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4 CYCLE
BRIGGS & STRATTON
GASOLINE ENGINES

Air-Cooled Power

of tools, dies and wear-resistant parts made from sintered and hot pressed carbides and cast nonferrous alloys. Also announced by Jessop Steel is the appointment of T. F. DeNormandie as branch manager of the Buffalo office at 1015 Liberty Bank Bldg. Mr. DeNormandie has been associated with the company for the past ten years, serving in various departments including production, sales, and the sheet and plate division.

♦

New offices have been opened by Bakelite Corp. in the Halliburton Bldg., Room 910, 1709 West Eighth St., Los Angeles 14.

♦

Bowser Inc. has named J. B. Trotman as head of the new Industrial Pump Division located in Fort Wayne, Ind. For the past six years Mr. Trotman served as general sales manager of Blackmer Pump Co., Grand Rapids, Mich.

♦

Recently announced by General Electric Co. is the appointment of Harold L. Aldrich as district representative in the New York office of the chemical department. In his new position Mr. Aldrich will serve as G-E sales representative for Glyptal alkyd resins.

♦

A new concern known as Holub Industries Inc. has been organized by Bert E. Holub, formerly general sales manager for Ideal Industries. The company has begun the manufacture of electrical and mechanical products in its new plant at 413 DeKalb Ave., Sycamore, Ill. Gordon W. Wetzel, previously Mr. Holub's assistant at Ideal, has been appointed sales manager.

♦

Having sold his interest in Ardee Plastics Inc., Bernard Schiller recently joined the firm of Banner Plastics Corp., 150 Bruckner Blvd., New York.

♦

Opening of a new branch office in the Claremount Hotel Bldg. in Evansville, Ind., has been announced by Allis-Chalmers Mfg. Co., Milwaukee. R. W. Gillmore, formerly sales engineer in the Indianapolis district office, has been appointed manager of the new office. Also announced by the company are improved testing and laboratory facilities for a variety of its industrial and marine products. Among these are a new "shock-test" laboratory developed in cooperation with the Navy during the war, a new electronic processing laboratory and a new steam turbine auxiliary test floor.

♦

Formerly Philadelphia branch manager, James Herman has been named eastern district sales manager for Ahlberg Bearing Co. of Chicago. His territory now covers Boston, New York, Philadelphia and Washington.

♦

John S. Barnes Corp. of Rockford, Ill., has placed E. C. Hawkins in charge of the recently opened Chicago branch office at 1112 Field Bldg., 135 South LaSalle St. Hydraulic sales in the Chicago area will be under the direct supervision of Mr. Hawkins, who for several years served as manager of the eastern sales office at Newark, N. J. Also announced by the company is the appointment of George M. Pearse Co., 965 Broad St., Newark 2, N. J., to serve as its sales rep-

precision PRODUCES PRECISION

TIMKEN BEARING EQUIPPED

IN THIS ULTRA-MODERN GEAR FINISHER

No spindle-operated machine tool can perform more precisely than its spindle bearings. The slightest inaccuracy in these bearings is sure to be reflected in the work.

Not even the least deviation of the cutting tool is permissible in this gear finisher, so the manufacturer, National Broach & Machine Co., Detroit, mounts the cutter spindle in Timken Zero Precision Bearings. Thus precision produces precision; gears finished in this machine consistently meet every test and tolerance.

Timken Tapered Roller Bearings are used at several other positions including the vertical and horizontal drive shafts and work centers.

To be sure of having Timken Bearing precision in your machines, make sure the trade-mark "TIMKEN" is stamped on the cup and cone of every bearing you use. The Timken Roller Bearing Company, Canton 6, Ohio.

TIMKEN

TAPERED ROLLER BEARINGS

● We have them every day, *but* we ask for them . . . precision parts have been our business for 44 years.

IGW has the experience, the facilities, and the craftsmen to produce—in either experimental or production quantities—parts in which the highest degree of precision performance is required.

Our reputation is your guarantee of protection against costly time and money losses due to rejections and scrap.

Indiana GEARS

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INDIANA GEAR WORKS • INDIANAPOLIS 7, IND.

resentative in the metropolitan New York and Newark territory. R. G. Pearse, hydraulic engineer, will supervise hydraulic sales in this territory.

♦

Expansion of its Vyncote Division has been announced by Watson-Standard Co. of Pittsburgh. This was planned to meet increasing demands for the company's series of vinyl plastic coatings for application in the metal products industry. The division now has independent technical and production personnel in a separate building with modern facilities for research and production.

♦

Roy A. Ekelhoff has been appointed sales representative for Titan Metal Mfg. Co. of Bellefonte, Pa., manufacturer of brass and bronze rods, forgings, die castings and welding rods. He will serve as representative in Iowa, Kansas, Missouri, southern Illinois and southwestern Indiana, and will make his headquarters at 817 Arcade Bldg., St. Louis.

♦

After four years in the Navy, Alfred Egerter has returned to the plastics industry. He has been named assistant sales manager of Paramet Corp., a subsidiary of Libbey-Owens-Ford Glass Co., Long Island City, N. Y.

♦

Opening of a new district office in the Paul Brown Bldg. in St. Louis has been announced by H. K. Porter Co. Inc. and Subsidiaries, Pittsburgh. Formerly assistant manager of the Porter Process Division in Pittsburgh, R. E. Nelson has been placed in charge of sales for Porter process equipment, Devine refinery and process equipment, Quimby pumps and Porter locomotives.

♦

Marine Products Co. has moved to new and enlarged quarters at 515 Lycaste, Detroit. The rapidly expanding sales and distribution division continues to be under the direction of Henry Clayton, sales manager.

♦

To succeed W. A. Taylor who resigned recently, A. M. Byers Co. has appointed George B. Coffey as manager of the Chicago district, with offices in the Conway building. The Chicago district includes the states of Michigan, Minnesota, North and South Dakota, Wisconsin, Iowa, Indiana, and parts of Illinois and northwestern Ohio.

♦

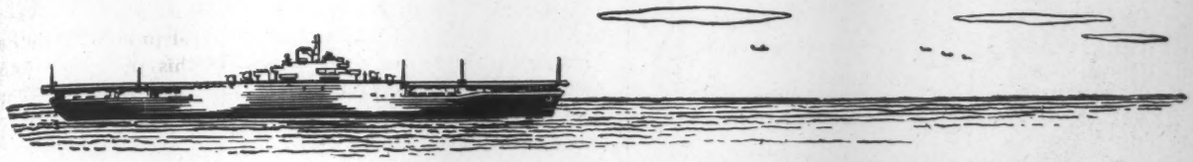
Election of Ralph R. Newquist as vice president in charge of sales has been announced by Roots-Connersville Blower Corp. of Connersville, Ind., one of the Dresser Industries. Mr. Newquist has been associated with Reliance Electric & Engineering Co., Louis Allis Co., and Allis-Chalmers Mfg. Co.

♦

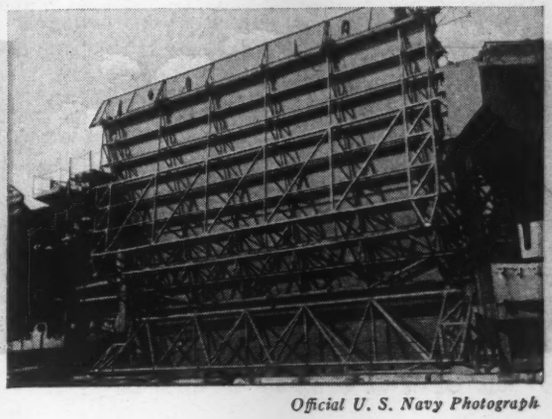
Formica Insulation Co. of Cincinnati has appointed the Gaines Hardwood Lumber Co. of St. Louis to serve as distributor of Formica decorative laminated plastics materials in the St. Louis area.

♦

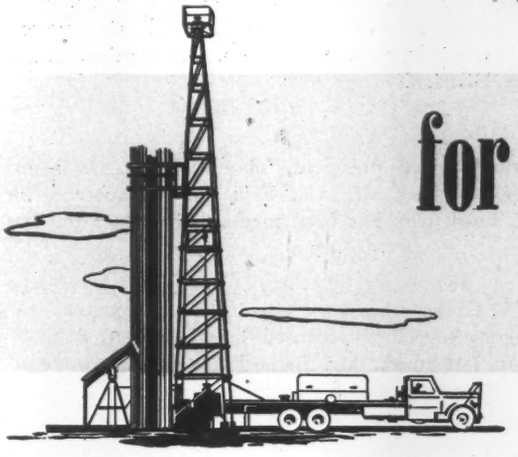
Experimental engineer for the last several years, William L. Jenkins has been made manager of the recently created electronic applications development department of The B. F.



War-born Ideas



Official U. S. Navy Photograph



for Peacetime Products:

lightweight structures with heavyweight strength . . . from B&W Tubes

For collapsible deck-edge elevators to whisk battle planes between hangars and flight decks of Essex class aircraft carriers, a lightweight yet exceptionally sturdy framework material was needed. The elevators overhang the deck without counterweighting, must support an 18,000 lb. load, absorb a terrific beating when submerged by heavy seas. B&W alloy tubing met the requirements with characteristic success.

There's an idea here for peacetime industry. B&W mechanical Tubing offers cost-cutting advantages, for instance, in lightweight, heavy-duty drilling rigs, in hoists and masts . . . telescoping or rigid. In dozens of other fields, too, B&W Tubing, developed for wartime jobs, merits investigation for weight-saving, money-saving use today.

B&W can supply mecharical tubing (either seamless or welded) of the proper analysis, size, gauge, temper, and finish for making practically any hollow part. Let us know what you plan to make. We may be able to suggest a B&W Tubing that's lighter, cheaper, and faster to fabricate than other materials you may have planned to use—and just as strong, too.

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Water-Tube Boilers, for Stationary Power Plants, for Marine Service • Water-Cooled Furnaces • Superheaters • Economizers • Air Heaters • Pulverized-Coal Equipment • Chain-Grate Stokers • Oil, Gas and Multi-fuel Burners • Refractories • Process Equipment.

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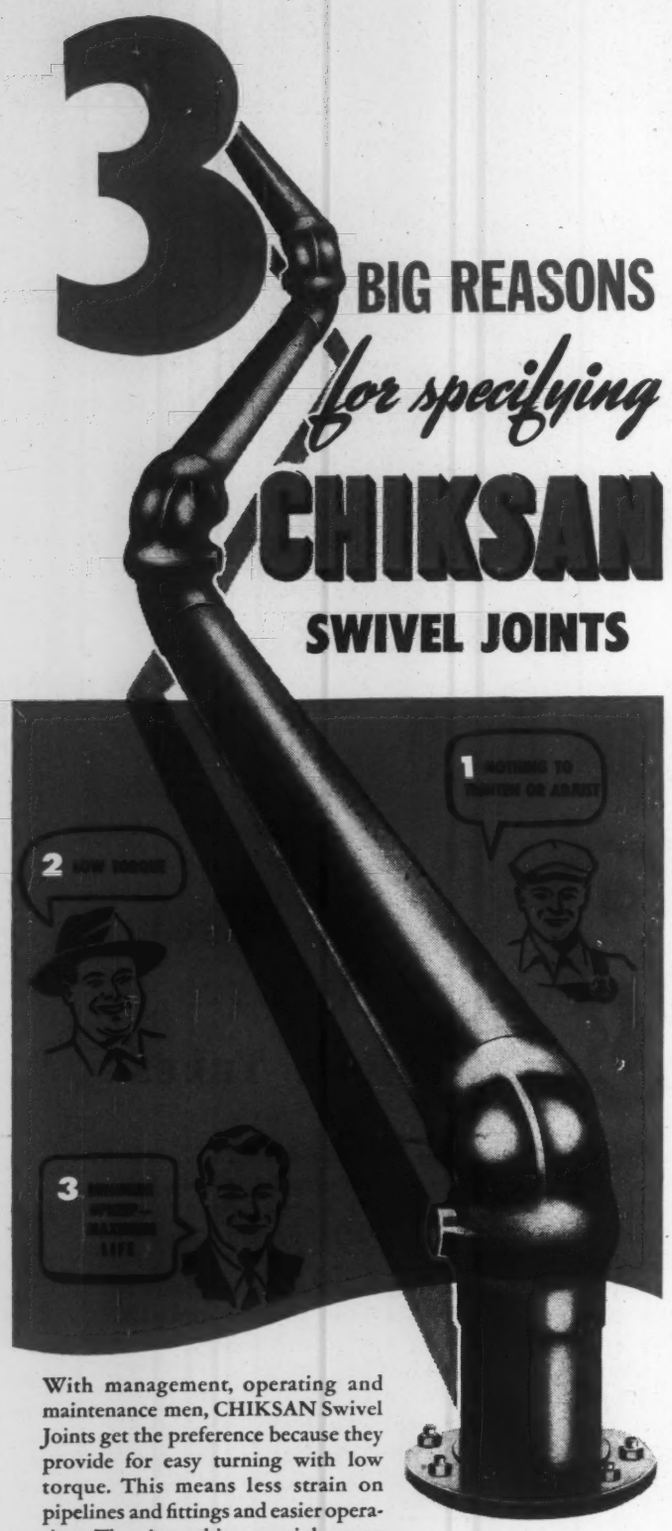
THE BABCOCK & WILCOX TUBE COMPANY

GENERAL OFFICES: BEAVER FALLS, PA.

PLANTS: ALLIANCE, OHIO AND BEAVER FALLS, PA.



With management, operating and maintenance men, CHIKSAN Swivel Joints get the preference because they provide for easy turning with low torque. This means less strain on pipelines and fittings and easier operation. There's nothing to tighten or adjust...and that means longer service with less maintenance attention and fewer shutdowns. Over 500 different Types, Styles and Sizes for pressures to 3,000 psi. and temperatures to 500° F., providing for full 360° rotation in 1, 2 and 3 planes. Write for latest catalog.

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New York 7 Houston 2

Goodrich Co. B. A. Evans, who served as technical superintendent of the metal products division and was superintendent of shops prior to his present appointment, has been named manager of the wheel and brake manufacturing department. The company recently purchased the airplane wheel and brake division of Hayes Industries Inc., where manufacturing operations are being conducted at present. Also announced by Goodrich is the appointment of R. J. Keller as technical manager of the wheel and brake manufacturing department.

♦

Arcos Corp., manufacturer of stainless and alloy electrodes, has opened a new district office at 450 Asylum St., Hartford 1, Conn., to serve the New England states. John K. Light, field engineer, is in charge of the new office and will be assisted by Charles B. Culbert. Both men served for a number of years in the home office in Philadelphia.

♦

Purchase of American Spiral Spring & Mfg. Co. of Pittsburgh has been announced by H. K. Porter Co. Inc., Pittsburgh. The newly acquired company, together with the present Porter Fort Pitt Spring plant, will be operated as American-Fort Pitt Spring Division.

♦

Formerly vice president, Thomas Lord has been elected president of Lord Mfg. Co., Erie, Pa. He succeeds his father, H. C. Lord, who has been made chairman of the board.

♦

C. V. Gregory has been appointed by The Reliance Electric & Engineering Co. to succeed Bon J. Ballard as district manager in Pittsburgh. Mr. Ballard has become assistant to the sales vice president and will continue to make his headquarters in Pittsburgh although his experience will be available to all sales districts of the company. L. J. Carr, a sales engineer in the Chicago office prior to two and a half years in the U. S. Naval Reserve, has joined the Reliance staff in Pittsburgh.

♦

Reorganization of the Tool & Die Division under the name of The Lester-Aetna Die Co. has been announced by Lester Engineering Co. The new organization, recently moved from Cleveland to Warren, O., offers complete engineering and manufacturing service for both die casting dies and plastic molds. It is jointly owned by Lester Engineering Co. of Cleveland and Aetna-Standard Engineering Co. of Warren, O.

♦

Connected with Lukens Steel Co. since 1937, H. K. Tryon has been made a member of the sales staff of the newly opened Cleveland office of the company and its subsidiary, By-Products Steel Corp. The new office is located at 1201 Citizens Bldg., Cleveland 14.

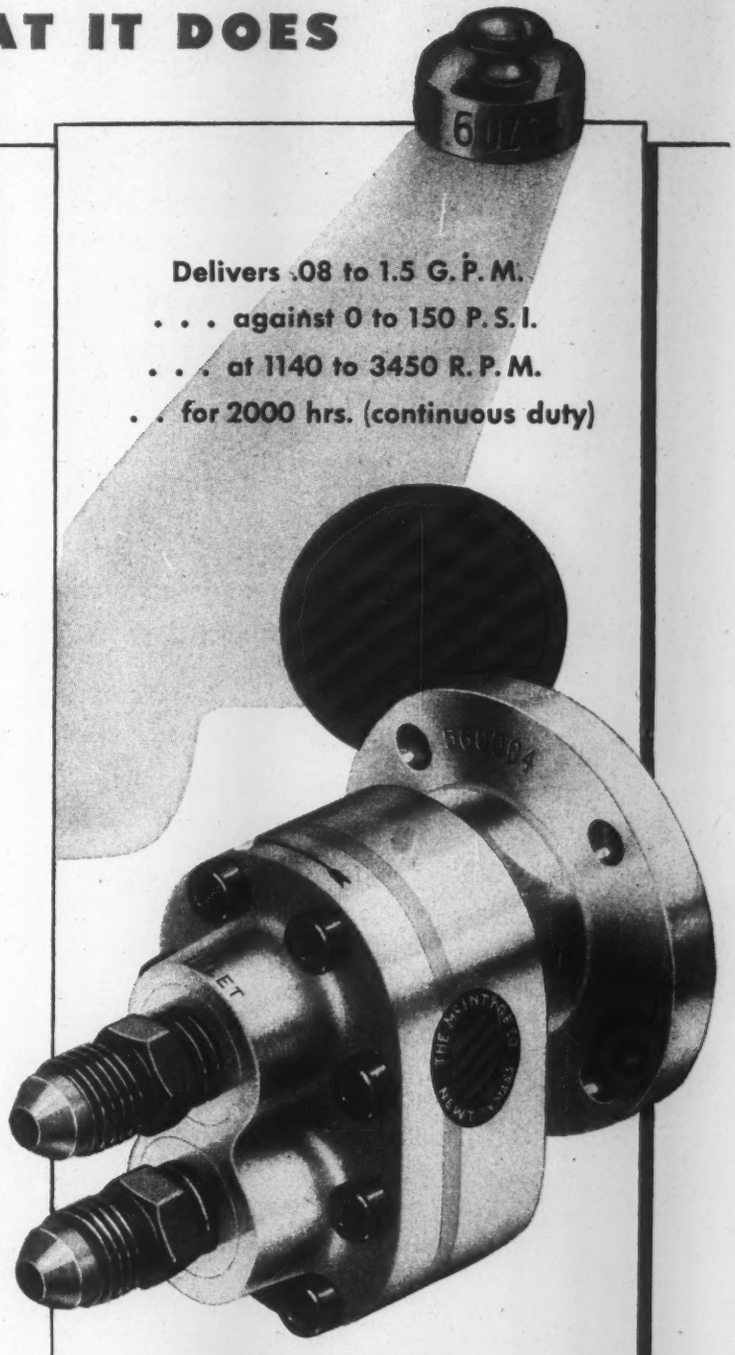
♦

Manufacturers Screw Products, Chicago, has changed its name to Stronghold Screw Products Inc. No change in management, personnel or products has resulted.

♦

Announced recently is the formation of the Parkway Foundry & Machine Corp. which occupies a new plant at 59 Paidge Ave., Brooklyn, N. Y. Nonferrous castings will be produced by the sand, permanent mold and centrifugal methods. Emile C. Mathis is president of the new company as well as the Matam Corp.

THIS PUMP WEIGHS ONLY 6 OZS. ...BUT SEE WHAT IT DOES


6 OZS.
Delivers .08 to 1.5 G. P. M.
. . . against 0 to 150 P. S. I.
. . . at 1140 to 3450 R. P. M.
. . for 2000 hrs. (continuous duty)
SCALE: ACTUAL SIZE
Available with 10,000 R.P.M. 1/20 H.P. Motor

If small size, light weight and low cost are important factors in your pump applications calling for low-capacity, low-pressure handling of oils, gasoline, alcohol, and certain mildly corrosive liquids, investigate McIntyre Series 1200 Pumps. You'll find that they actually cost less than larger, heavier pumps made to less-exacting specifications.

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Because McIntyre machining methods are *capable of making surfaces flat to one light band . . . and holding vital dimensions to tolerances of tenths and split tenths,* the close fit of gears, center plates and sides of McIntyre Pumps assures high volumetric efficiency. That is the plus the McIntyre light-band trade-mark promises.

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also makers of precision spur gears to your specifications.

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PUMPS AND FLUID MOTORS

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Meetings and Expositions

Sept. 16-20—

Instrument Society of America. First national instrument conference and exhibit to be held at William Penn Hotel, Pittsburgh. Richard Rimbach, 1117 Wolfendale St., Pittsburgh 12, is executive secretary.

Sept. 18-20—

Illuminating Engineering Society. Victory convention to be held at Chateau Frontenac, Quebec, Canada. A. D. Hinckley, 51 Madison Ave., New York 10, is executive secretary.

Sept. 30-Oct. 2—

American Society of Mechanical Engineers. Fall meeting to be held at Boston. Additional information may be obtained from headquarters of the society at 29 West 39th St., New York. C. E. Davies is secretary.

Oct. 3-5—

National Electronics Conference Inc. National forum on electronic research, development and application, and exhibits of electronic equipment to be held at Edgewater Beach Hotel, Chicago 14. E. H. Schulz, Illinois Institute of Technology, Technology Center, Chicago 16, is secretary.

Oct. 3-5—

Society of Automotive Engineers Inc. National aeronautical meeting and aircraft engineering display to be held at Biltmore Hotel, Los Angeles. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Oct. 9-11—

Porcelain Enamel Institute. Eighth annual forum to be held at University of Illinois. Additional information may be obtained from headquarters of the society at 1010 Vermont Ave., N.W., Washington 5.

Oct. 10-12—

American Society of Tool Engineers. Semiannual national convention to be held at Hotel William Penn, Pittsburgh. Additional information may be obtained from headquarters of the society at 1666 Penobscot Bldg., Detroit 26.

Oct. 16-17—

Society of Automotive Engineers Inc. National transportation and maintenance meeting to be held at Hotel Knickerbocker, Chicago. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Oct. 23-25—

American Society of Body Engineers Inc. Technical convention to be held at Rackham Memorial Bldg., Detroit. Otto Graebner, Murray Corp., Detroit, is secretary.

Oct. 28-30—

American Gear Manufacturers Association. Semiannual meeting to be held at Edgewater Beach Hotel, Chicago 14. Newbold C. Goin, Empire Bldg., Pittsburgh 22, is executive secretary.

Nov. 7-8—

Society of Automotive Engineers Inc. Fuels and lubricants meeting to be held at Mayo Hotel, Tulsa, Okla. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Nov. 18-22—

National Metal Congress and Exposition to be held at Municipal Auditorium, Atlantic City, N. J., in conjunction with the annual meetings of the following societies: American Society for Metals; Iron and Steel Division and Institute of Metals Division of American Institute of Mining and Metallurgical Engineers; American Welding Society; and American Industrial Radium and X-Ray Society. Chester L. Wells, 7301 Euclid Ave., Cleveland 3, is assistant managing director of the exposition.